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THE SALVAGE OF THE U. S. S. *SQUALUS*.

BY LIEUTENANT COMMANDER FLOYD A. TUSLER, (CC),
U. S. NAVY, MEMBER.*

The author served as Salvage Officer on the staff of the Commander Squalus Salvage Unit. He presents the steps undertaken in raising the submarine Squalus. The Society welcomes the opportunity to publish these particulars, should they be of assistance to any navy.

The U. S. S. *Squalus* was completed by the Navy Yard, Portsmouth, N. H., on 12 May 1939. From then until the 23rd of May she was engaged in training operations in preparation for her preliminary trials, scheduled for the second week in June. At 0740 on the 23rd of May, while making a quick dive, she sank in 240 feet of water about five miles south of the Isles of Shoals off Portsmouth

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Harbor. On this dive, the main engine air induction outboard valve and the four hull valves at the inboard end of the pipe lines, connecting the outboard valve with the after battery space and two engine rooms, were all open. These spaces were quickly flooded, and, as learned later, due to an open bulkhead door, the after torpedo room was also flooded. By prompt action, the watertight closures in the after bulkhead of the control room were closed, thus preventing water, except in small quantity, from entering the three forward hull compartments.

Thirty-three of her crew, including four officers and one civilian, were rescued from her by the U. S. S. *Falcon,* submarine rescue vessel, in four trips of a rescue chamber to the forward torpedo room escape hatch on the 24th of May. There was every reason to believe that no one was alive in the after part of the boat, but, to make sure, a fifth trip of the rescue chamber was made on 25 May to the after torpedo room escape hatch, which definitely established that the after torpedo room, too, was completely flooded. This concluded the rescue phase of operations.

That night Rear Admiral C. W. Cole, U. S. N., who had been designated Commander Salvage Unit, held a conference in his office at the Portsmouth Navy Yard, at which a tentative salvage plan was adopted and the Salvage Force organized. Salvage operations began the next day and were prosecuted work days, Sundays, and holidays, weather permitting, sometimes late into the night.

It was evident, even during the rescue work, that too much reliance could not be placed on divers working at depths below 200 feet when breathing air. In general, a diver became more or less punch drunk or slap happy by the time he reached the bottom, having many of the symptoms of an intoxicated man. Divers had so much difficulty in coordinating their efforts that, even after careful rehearsal of the work on the *Sculpin,* sister ship of the *Squalus,* several dives would sometimes be required for a simple task that would take only a few seconds on the surface. This condition was somewhat aggravated at the start by not using sufficient air pressure on the diver's air line, which increased the carbon dioxide concentration in the suit. Beginning on the 29th of May, the pressure on the diver's air line was increased from 185 to 250 pounds per square inch, resulting in a marked improvement in diving conditions.

The Experimental Diving Unit, whose primary mission is furthering the art of diving, had done a great deal of research work on the use of a mixture of oxygen and helium in place of air for diving operations. During the two previous summers the Unit worked with this mixture in various depths of water up to 400 feet off Portsmouth Harbor and decompression tables had been worked out for divers using the mixture. It was found that, when breathing a mixture of oxygen and helium, a diver's mind was much clearer than if he breathed air; hence he could be relied upon to do more complicated work, although it is doubtful that he had any more physical capacity, the work being equally tiring on air or the mixture. In connection with this research work, an experimental helmet had been devised which permitted recirculation of the mixture in the diver's suit through a canister containing a carbon dioxide absorbent, thus materially reducing the amount of the mixture that had to be supplied the diver. As might be expected, this helmet was not very well suited to practical diving conditions. Some changes were made at the very start of the *Squalus* salvage operations, so that almost from the beginning, practically all dives in deep water were made on the oxygen-helium mixture. Certain unsatisfactory conditions, such as insufficient recirculation, insufficient CO_2 absorbent, too much noise, fogging up of face plates, awkward projections from the helmet, etc., were gradually eliminated. Electrically heated underwear was obtained to overcome the undue cooling effect of the mixture on the diver and the old batteryless telephone was replaced by a modern vacuum-tube outfit, which greatly improved communication with the diver.

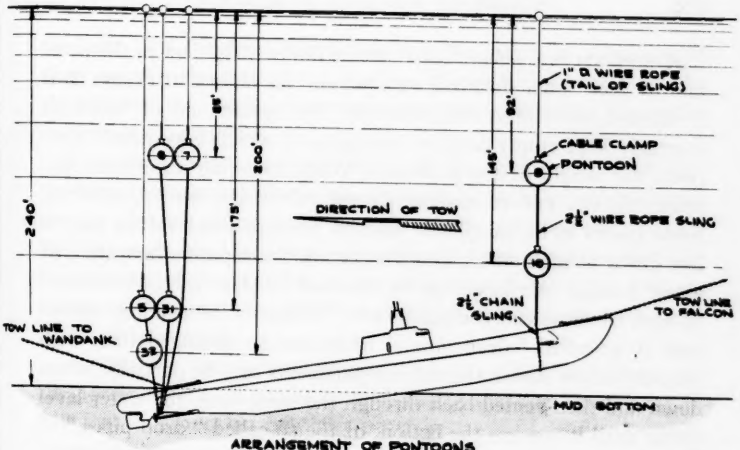
For a short time, while certain changes were being made to the recirculating type of helmet, recourse was had to the use of the oxygen-helium mixture in the ordinary air suit. The consumption of the mixture was, however, so great as to make this almost prohibitive, several bottles of gas being required for a single dive.

Regardless of what salvage plan was to be used, certain factors had to be taken into consideration. First, it is impossible, by the use of submersible pontoons and a vessel's own buoyancy, to lift both ends of a sunken vessel simultaneously, because the weight and center of gravity of the water in flooded compartments is not accurately known and because the amount of mud suction, which

may amount to a hundred tons or more, is unknown. Hence, one end or the other always comes up first. The salvage force can, of course, so plan the blowing operations that the desired end rises first. Any attempt to raise one end of the *Squalus* from a depth of 240 feet would have stood her almost on end. At this angle, practically all of the air would have been spilled from any of her ballast tanks that had been blown dry, these would reflood, and the upraised end would then sink back to the bottom. Thus, it was necessary to lift the *Squalus* in two or more stages. By making the lift in three stages, the angle attained by the vessel with one end up would be limited to about 20 degrees, at which angle the loss of air and consequent reflooding would not be excessive. Hence, it was decided to lift the *Squalus* in three stages of about 80 feet each. After the first lift of 80 feet she was to be towed into shallower water until she grounded. This procedure was to be repeated until on the third, and last, lift she would draw not over 40 feet of water, the maximum draft that would allow her to be towed to the Navy Yard, Portsmouth, N. H.

Since a pontoon ceases to rise after it has surfaced, any further upward motion of the vessel being raised tends to reduce the lift exerted by the pontoon until eventually, if the vessel continues to rise, the loss in lift is equal to the total lift originally exerted by the pontoon. If enough pontoons are so arranged at the same upper level that they surface simultaneously, the loss in lift can be made sufficient to overcome any additional buoyancy gained by the vessel on her way up due to the expansion of air in partially blown tanks, stop the momentum acquired by the vessel, and counteract the additional initial lift necessary to break any mud suction. Under these circumstances, the vessel would be lifted upward a distance equal to the depth below the surface at which the upper level of pontoons are set and she would, of course, remain suspended from them. At no time during the lift can either end of the vessel herself be permitted to have positive buoyancy, otherwise it will continue to rise and resink as explained above.

Without unwatering any of the four flooded compartments aft, but by blowing empty all of the main ballast tanks and fuel oil tanks, calculations indicated that five pontoons would be required at the stern and two at the bow. These were to be arranged as shown in Figure 1.



ARRANGEMENT OF PONTOONS
LIFT OF 13 JULY, 1939

FIG. 1.

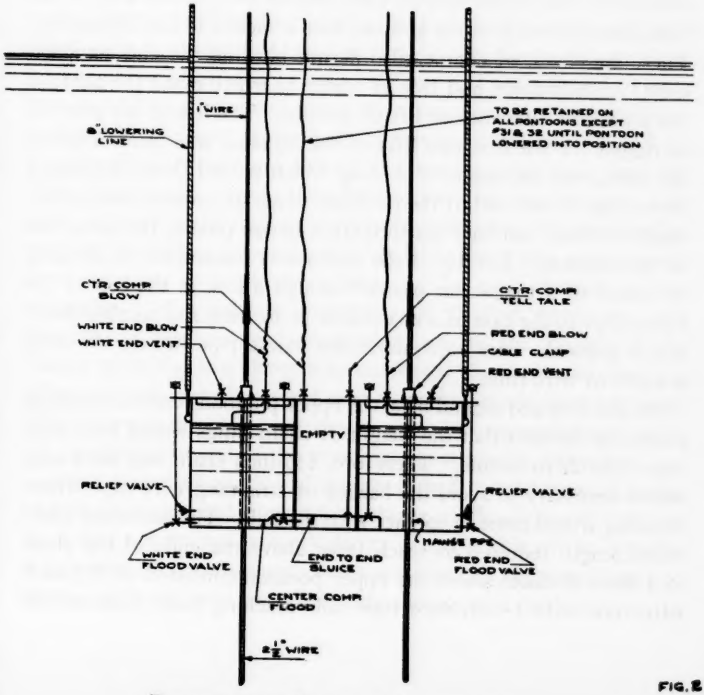


FIG. 2

PONTOON WITH GEAR AND APPURTENANCES.

A pontoon is a submersible steel cylinder 13½ feet in diameter and 32 feet long, sheathed with wood. As shown in Figure 2, it is divided into three compartments, the center one of which is normally kept empty and the buoyancy of which very nearly supports the weight of the pontoon. When lowering a pontoon into place, the two end compartment vent valves are opened, allowing these spaces to be completely flooded through flood valves located low down at each end, giving the pontoon a negative buoyancy of about 4 tons. Air hoses can be attached to blow valves connected to each of these end compartments. These air connections terminate in so-called "drop pipes," which are so arranged that, with the vent valves closed, the end compartments may be partially blown down and then vented back through the same hose, the water level rising until it reaches the bottom of the so-called "drop pipes," at which point further reflooding of the end compartments will be prevented. In this condition the pontoon has a positive buoyancy of about 8 tons. This is done when setting the pontoon after it has been lowered into position and has been attached to its lifting sling. These air hoses, of course, also permit blowing the end compartments entirely down and free of water, in which event the pontoon has a positive buoyancy or lift of 80 tons. The lift of the pontoon as rigged for the first two lifts of the *Squalus* was transmitted to the submarine by means of a sling which passed down through a hawse pipe at one end of the pontoon, down to, around, and underneath the hull, and back up through a hawse pipe at the other end of the pontoon. The lift of the pontoon is transmitted to its sling by means of a toggle bar passed through a link at the top of the hawse pipe in the case of a chain sling or by means of a cable clamp which grips the wire rope above the hawse pipe in case the sling is made of wire rope.

On the first and second lifts, the upper pontoons were set so high above the *Squalus* that the weight of chain slings would have been very difficult to handle. Therefore, 2½-inch chain was used only where necessary to avoid the hazard of stranding wire rope where the sling would come in contact with the hull. The necessary additional length required to reach from above the ends of the chain to a short distance above the upper pontoon consisted of 2½-inch wire rope with 1-inch wire rope tails reaching from there to the

surface. On the final lift, for which the pontoons were rigged close down to the deck of the *Squalus*, 2½-inch chain was used for the entire sling, with 1-inch wire rope tails, as before, from the pontoon to the surface.

The salvage work obviously called for the attachment of salvage air hose to the main ballast tanks and fuel oil tanks and necessitated getting pontoon slings under the hull in such positions that, when a lift was made, they would not slip off. As there would be no point in attaching air hoses unless a reeving line for hauling the pontoon slings could be passed under the submarine, this was, of course, the first task to attempt.

At the time that salvage operations began, little was known of the actual conditions on the bottom, except that the *Squalus* lay in 240 feet of water, her bow was up at an angle of 10 degrees, and she was about on an even keel athwartships. As the keel of a submarine slopes up considerably at each end, it was anticipated that a diver could drop to the bottom, walk under her stern, and pass a line to another diver on the bottom on the other side provided with a similar line, who would tie the lower ends of these lines together, thus making a reeving line. It was desired to have the after reeving line, with which the after pontoon slings would be hauled under, placed between the propeller shafts and the hull so that the slings would not slip off the stern of the boat when the stern was raised. It was thought that the diver would have difficulty in reaching up high enough from the bottom to pass the line between the hull and the propeller shafts. There was some worry, also, about the diver dropping down so far over the side of the submarine to the bottom, a distance of 22 feet, hence you can imagine the surprise and disappointment on the fantail of the *Falcon* when the first diver who attempted this, reported over the telephone that he had stepped over the side directly into soft mud. The stern was down in mud almost up to her superstructure deck. the extreme after end of the vessel being completely buried.

A self-propelled nozzle had already been devised for washing out tunnels and was used during the salvage of the *S-51* and the *S-4*. As it was not desired to work divers over the side of the *Squalus* at this depth unless absolutely necessary, the idea suggested itself of using a sectional pipe lance bent into the arc of a circle of just the

right diameter to pass around the hull inside of the propeller shafts. It was hoped that, in the relatively soft mud, a diver might be able to push the lance down and around under the hull from the deck of the submarine, disconnecting the fire hose and adding on new sections of pipe as the lance was advanced. Such a lance was made up and provided with a self-propelled nozzle, the sections of pipe being joined by ordinary pipe unions.

After some difficulty, the nozzle of the lance was advanced down, under the keel, and up a short distance on the other side, but as soon as the nozzle was directed upward on the other side, the water no longer flowed back along the down-going part of the lance and it soon became stuck fast in the mud. Several attempts were made to force a flexible steel rod or "snake" down through the lance, out of the nozzle, and up through the mud on the other side, but the mud was too hard for this and, as it was probable the pipe unions had loosened up, there was a possibility that the lance had departed from its original circular shape and had become more or less spiral-shaped, so that the nozzle was no longer headed in the right direction.

Resort was next had to the customary method of getting a reeving line under a vessel, which is to wash out a tunnel under the hull with a fire hose and then pass a reeving line through the tunnel. This was tried and abandoned as too dangerous at this depth after one attempt was made in which the diver overworked himself, passed out on the bottom, and had to be hauled up to the surface without the usual decompression on the way up, as a result of which he suffered a case of "bends."

As a last resort, pontoons could have been rigged at the bow, which was clear of the mud for a distance of about 125 feet, then the bow lifted high enough with the pontoons to permit sweeping a wire all the way under the boat from the bow back under the stern. This would, however, have placed the reeving wire outside of the propeller shafts. With the pontoon slings in this position, the shafts would have been hopelessly ruined and there was some danger that the bending of the propeller shafts would open up a leak into No. 4 main ballast tank or the after torpedo room.

As tunnelling seemed to be out of the question, the obvious solution seemed to be a circular, sectional, pipe lance, provided with a

self-propelled nozzle as before, but having telescopic, self-aligning joints of such a simple design that a diver could easily make them up on the deck of the submarine and having joints designed to prevent one section from turning relative to the adjoining ones. Such a lance, the general features of which are shown in Figure 3, was soon made up by the Navy Yard. The first section was 15 feet long; each of the other sections was 8 feet long. Holes were drilled in the lance every 4 feet along its length and were so ar-

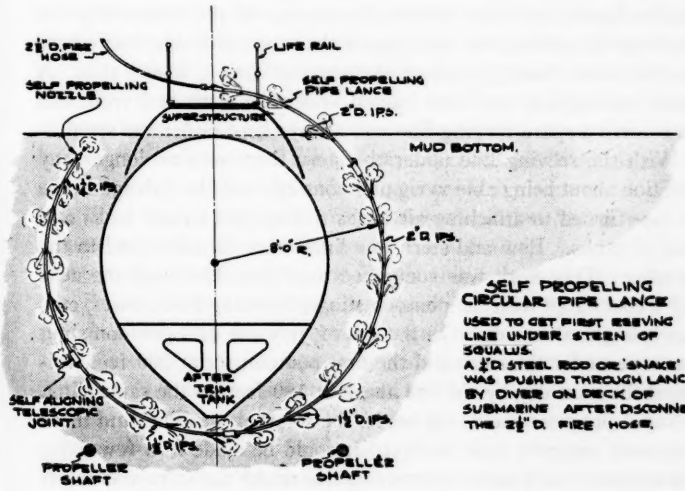

2½" D. FIRE HOSE
LIFE RAIL
SELF PROPELLING NOZZLES
SELF PROPELLING PIPE LANCE
SUPERSTRUCTURE
2" D. IPS.
MUD BOTTOM.
½" D. IPS.
8'-0" R.
2" D. IPS.
AFTER TRIM TANK
SELF ALIGNING TELESCOPIC JOINT.
1½" D. IPS.
1½" D. IPS.
PROPELLER SHAFT
PROPELLER SHAFT
SELF PROPELLING CIRCULAR PIPE LANCE
USED TO GET FIRST RESEVING LINE UNDER STERN OF SQUALUS.
A ¼" D. STEEL ROD OR 'SNAKE' WAS PUSHED THROUGH LANCE BY DIVER ON DECK OF SUBMARINE AFTER DISCONNECTING THE 2½" D. FIRE HOSE.

FIG. 3

ranged that when water was forced out of them it was directed in a backward direction, the reaction of which tended to force the lance through the mud and at the same time the mud was kept washed away from around the lance. The sections of the pipe lance were graduated in diameter from 2 inches for the section next to the hose to 1 inch for the nozzle section in order to provide a uniform supply of water along the whole length of the lance.

This lance proved to be entirely successful and was worked down, around the hull, and up on the other side, the fire hose being disconnected and new sections added on as necessary until the nozzle was only a couple of feet below the mud level on the other side.

Rather than add on a new section, the fire hose was disconnected, and a ¼-inch steel "snake," having a 3/16-inch steel wire rope reeving line attached to its rear end, was pushed through the lance and up through the mud on the other side. This was soon located by a diver, who attached a line to it that was sent down from the *Falcon,* and the "snake" was hauled up on the 21st of June, Admiral Cole's birthday. A heavier reeving wire was then spliced onto the down-going end of the first one, and so on until a 1-inch steel wire rope was under the stern. The lance itself was, of course, hauled up by the reeving line as soon as the diameter of the down-going reeving line was greater than the inside diameter of the nozzle, a stop being placed on the reeving wire to insure this. A spare reeving line was next hauled under the stern and from this time forth a spare reeving line was always kept under the stern.

With the reeving line under the stern there was no longer any question about being able to rig pontoons aft, and the Salvage Force at once turned to attaching air hoses to the main ballast tanks and fuel oil tanks. Bow and stern tow lines were then attached to the *Squalus.* This work was completed on 2 July 1939 and the next day a coal barge with the pontoon slings, lowering lines, hoses, etc., was brought alongside. On the 4th of July the first pontoon sling was rove under the stern and the first pontoon set at 200 feet, followed the next day by the one above at 180 feet on the same sling.

Rapid progress was being made, the weather was fine, and there was every prospect that the first lift could be made in a few days. The regular 1-inch wire rope reeving line under the stern was taken in hand, preparatory to hauling under the next sling, but it would not reeve; neither would the spare 7/16-inch wire rope reeving line, which, due to slack, had become hopelessly fouled on the deck of the *Squalus.* The 1-inch reeving line had become caught between the sling of the two pontoons just set and the hull. These pontoons were reluctantly unrigged. This freed the 1-inch reeving line and both the other slings were then hauled under the stern by it at one time without further trouble. The five pontoons at the stern were rigged and set in two days, the only untoward incident occurring just after the fourth pontoon had been lowered into place at a depth of 80 feet. The diver, in coming up from closing the vents in the end compartments accidentally tripped the pelican

hook by which one of the lowering lines was attached to one end of the pontoon. The entire weight of the pontoon was thrown on the lowering line at the other end and this, plus the heaving of the *Falcon* in the sea against the 120-ton inertia of the pontoon, broke the pelican hook attaching this lowering line to the pontoon, allowing the pontoon to drop down to a depth of about 150 feet on top of another pontoon that had already been set on that sling. As soon as the diver was clear, the pontoon was placed on positive buoyancy, the cable clamps holding it down to that depth. It then being late at night, the pontoon was left to ride that way until the next morning, when new lowering lines were attached and the pontoon was raised up and set at its desired depth of 80 feet.

In the meantime, the Diesel oil in the fuel oil tanks had been siphoned out of the *Squalus* through the regular salvage air hoses into the fuel oil tanks of the *Falcon,* the Diesel oil being automatically replaced by salt water via the compensating water line. This was done to avoid contaminating the local beaches with Diesel oil when these tanks were blown empty the first time.

After rigging and setting the fifth, and last pontoon at the stern, the *Falcon* shifted moorings to a position over the bow of the *Squalus,* the sling for the bow pontoons was rove under the bow with a reeving wire that had previously been swept under the bow. The two bow pontoons were rigged on this sling on the 12th day of July and the *Falcon* shifted moorings clear of the *Squalus* in anticipation of making the first lift next day. During the night, the air hoses to the pontoons and submarine were connected to the air manifold on the *Falcon* and checked, during which it was discovered that one of the most reliable divers had inadvertently closed the air salvage valve on the submarine to No. 2 main ballast tank instead of the air salvage valve to the control room, which was quite close to it. This incident is typical of many similar ones that happened before the *Squalus* was brought into shallower water and is only mentioned to emphasize how easy it was for divers to make mistakes at this depth of water, where the pressure ran up to 105 pounds per square inch. Handicapped also by a clumsy diving suit and a maze of air hoses and descending lines, it is not remarkable that on some days, after seven or eight divers had done their utmost, very little, if any, progress was made. The above error

was rectified early next morning before any attempt was made to raise the *Squalus*.

In the meantime, during the night, all of the upper pontoons and certain of the fuel oil tanks had been blown empty, the calculations having shown that this could be done without danger of prematurely raising the submarine.

Early on the 13th of July, the tug *Wandank* picked up the tow line attached to the stern of the *Squalus*, it having been decided to tow her stern first, as she had sunk with her stern headed in the desired direction toward shallow water, and to have turned her around would have risked fouling some of the *Falcon's* moorings.

The *Falcon* was to be towed by a tow line attached to the bow of the *Squalus*. This tow line was picked up by the tug *Sagamore*, preparatory to passing it to the *Falcon* after the *Squalus* was lifted, but before the tow got underway.

All preparations having been completed for making the lift, the final blow was started at 0930. The blowing procedure called for lifting the stern first, because the cutaway at the bow provided a more suitable pivoting point than the rudder and gave the submarine more stability during the lift, due to the fact that the upward reaction of the bottom acted at a point higher above the keel than would have been the case with the bow up and the stern supported from the bottom of the rudder. Accordingly, main ballast tanks and fuel oil tanks were blown in succession, starting from aft. It was not until all of the after group of main ballast tanks and all but one of the fuel oil tanks had been blown and while blowing No. 1 fuel oil tank that the stern rose at 1450, preceded and accompanied by a huge boil of air, and surfacing pontoons Nos. 6 and 7. The slings of these two pontoons went slack, allowing the pontoons to be swept some distance apart, showing that the stern was light; but, as the stern rose higher and the angle of trim increased, air was spilled from tanks already blown and these reflooded enough to give the stern sufficient negative buoyancy to arrest its upward motion. The stern then sank until its negative buoyancy was taken by the slings of pontoons Nos. 6 and 7, which brought them back together alongside one another on the surface.

As soon as these two pontoons had been secured by closing their blow and flood valves, blowing was continued to raise the bow.

The vessel had proved to be considerably heavier than anticipated, with the result that, when the time came to raise the bow, only No. 1 and No. 2 main ballast tanks remained unblown. No. 1 main ballast tank was relatively small and, should the bow start up with it only partially blown, the surfacing of the upper pontoon at the bow could be expected to counteract any additional buoyancy acquired by this tank due to the expansion of air in it during the ascent and to counteract the upward momentum acquired by the bow. When the stern had lifted, the ship, of course, had pivoted and the bow had sunk down an unknown amount into the soft mud of the bottom, but, due to the shape of the bow, it was thought that the mud suction would be relatively insignificant. Consequently, No. 1 main ballast tank was blown next, but this failed to raise the bow.

It had been planned to rig each of the bow pontoons on a separate sling of 2¼-inch galvanized wire rope, since there was not enough 2½-inch plow steel wire rope available at this time to make a sling for the bow, but, at the last moment, in the interest of time-saving and as it was possible that the lift of the lower pontoon would not be required, the two bow pontoons had been rigged on the 2¼-inch galvanized wire rope sling originally intended for only the upper pontoon. With both pontoons fully blown, the strength of this sling would be taxed to the limit, hence it seemed better to blow No. 2 main ballast tank next, rather than the lower pontoon at the bow, with the hope that the bow would not start up until this tank was almost fully blown.

At 1612, with No. 2 main ballast tank only about one-third blown, the bow started up, the upper pontoon surfaced, followed very shortly by the lower pontoon, and then very soon by about 20 feet of the bow of the *Squalus*. About this time the sling, by which the stern pontoons No. 5 and No. 32 were attached to the submarine, broke. These two pontoons surfaced and drifted off. Stern pontoons Nos. 6 and 7 were pulled under and the vessel sank back to the bottom stern first at an angle of about 60 degrees, the main ballast tanks spilling most of their air at this angle and reflooding, so that the bow again became heavy and sank. On the way up, the bow had slipped out of the sling of the two bow pontoons. Some doubt had existed whether this sling was abaft both of the rigged-

out bow planes when the pontoons were set. It was now quite definitely indicated that the sling had been behind only one of the bow planes, otherwise the bow could hardly have slipped out of the sling. The two bow pontoons were, however, still anchored to the submarine, because, even though their sling had slipped off the bow, it was still underneath the bow tow line that had been picked up by the *Sagamore*.

The disastrous surfacing of the bow may be explained as follows: First, there was an unknown amount of mud suction tending to hold the bow down, which was suddenly released when the bow started up. Second, there was more or less free water in the three forward hull compartments which ran aft as the bow lifted, tending to make the bow lighter and lighter. Third, the air in No. 2 main ballast tank and in the lower pontoon at the bow, both of which were only partially blown, expanded as the bow rose and the sea pressure became less and less, thus blowing more and more water from those spaces and increasing the buoyancy at the bow. Fourth, the momentum attained by the bow tended to carry it still higher, even after a condition of static equilibrium had been established between the weight and buoyancy of the bow. The combined effect of these four factors was not only greater than the stopping effect (loss of buoyancy) due to the surfacing of the single upper pontoon, but sufficient also to outweigh the small additional stopping effect due to the surfacing of the second (partially blown) pontoon, with the result that the bow of the submarine itself actually attained positive buoyancy for a time. A further contributory factor was the buoyancy gained by ballast tanks that were "fully-blown" with the stern up. Actually, such tanks could only be blown down to the highest point of any flood valve opening and, with the stern up, all of their buoyancy was not recoverable, but, as the bow rose, thus leveling off the ship as a whole, and the air inside expanded due to the lessened sea pressure, these tanks gained more and more buoyancy until all their buoyancy had been fully recovered when the ship had leveled off. This process was, of course, reversed as the bow continued to rise after the ship had leveled off, air tended to spill out of such tanks, and reflooding took place at an increasingly greater rate as the bow rose higher and higher, thus explaining why the bow did not stay up after it had surfaced.

The two stern pontoons, Nos. 5 and 32, that broke their sling, drifted off, the lower one in a sinking condition, due to its having struck another pontoon on the way up. Both, however, were secured and later unrigged and towed back to the Navy Yard for repairs. The sling of the two bow pontoons had fouled a number of air hoses attached to the *Squalus* and, as the weather turned bad on the afternoon of the lift, it was several days before these two pontoons could be unrigged and cleared from the submarine and returned to the Navy Yard for repairs, the red end of the upper pontoon having been struck by the lower pontoon when it surfaced.

Three pontoons, Nos. 6, 7, and 31, were still attached to the stern of the *Squalus* when it sank back to the bottom. The condition of these pontoons and their slings was unknown, hence, it was decided to unrig them completely and pull out their slings for inspection. As will be seen, this proved to be a most discouraging task. The eye had pulled out of one of the buoys supporting the tail-end of the sling of one of these pontoons during the lift, allowing the 1-inch wire rope tail to drop down on top of the pontoons, where it became hopelessly fouled with air hoses, valves, cable clamps, etc. The ends of the other three wire rope tails were still supported by buoys on the surface, but had become entangled by the swirl accompanying the rise of the stern. The air hoses to the two pontoons at the stern which broke their sling had, of course, been cast off from the pontoons and, as these hoses were entangled with the air hoses to the three pontoons still attached to the stern, they added still further to the difficulties. Moreover, most of the air hoses still attached to the stern pontoon were ruptured and had to be renewed.

As pontoons Nos. 6 and 7 had been pulled under while on full buoyancy with blow and flood valves closed, the blow valves had to be opened, sea pressure built up with air in the end compartments, the flood valves opened, and the end compartments vented back to allow them to flood up to the bottom of the drop pipes to place the pontoons back in a safe condition to ride on their slings with a positive buoyancy of about 8 tons until such time as they could be removed.

The bow tow line, consisting of about 150 feet of ¾-inch wire rope attached to the bow and 120 fathoms of 7-inch Manila line

on the end of this, had been picked up by the bight of the sling of the two bow pontoons as it slipped off the bow and was carried aft on the submarine, where it became entangled with the air hoses and upper works from the bow to the conning tower fairwater with several long bights of 7-inch Manila line over the side.

It took until the 2nd of August to clear up the wreckage caused by the lift of 13 July. The removal of the pontoon No. 31 from the stern was particularly trying. This pontoon had originally been set at 200 feet below the surface. Four dives were made down to it on 26 July to inspect it and clear away small wires and lines fouled around it. The pontoon was now at a depth of about 225 feet and up at the white end at an angle of about 45 degrees. Apparently one or more of the compartments had leaked enough to give the pontoon negative buoyancy and cause it to sink to the bottom, with the white end resting on the stern of the *Squalus*. On the 27th, ten dives were made to this pontoon; two divers attached 1-inch wire rope lifting lines to the red and white ends, two divers accomplished nothing due to ear trouble, and the remaining six divers were employed in clearing turns taken by the red-end lifting wire around the 2½-inch wire rope sling at the red end and in inspecting the bridles of the cable clamps to determine the work necessary to prepare the cable clamps for lifting, these bridles evidently having been damaged when the pontoon above had been accidentally dropped on this one while it was being set.

Only three dives were made to the pontoon on the 28th and, due to the rolling of the *Falcon*, poor visibility, and motion of the pontoons, nothing useful was accomplished. Three dives were made on the 29th. The first diver attached a lifting line to the bridle of the cable clamp at the white end, noting that one of the wedge chains was broken. The second diver found that a brass disc from the cable clamp of the pontoon above had slid down the sling on top of the red-end cable clamp, thus preventing him from attaching a lifting line to this cable clamp. The third diver repaired the broken wedge chain of the white-end cable clamp with seizing wire and opened the vent valve to the white-end compartment to make that end heavy. The white-end cable clamp was then hauled up and, by hauling up on the red end of the sling, the sling was hauled out from under stern and unrove from the hawse pipes of the pontoon,

the red-end cable clamp being brought up by the 2½-inch chain, since the chain was too large to pass through the cable clamp.

On the 30th, an attempt was made to lift the pontoon, the red end was lifted about 20 feet to level off the pontoon, but, when an attempt was made to lift the white end, the long link, to which the lifting line was attached, parted. The red end was then lowered to the bottom and a diver sent down to shackle the white-end raising line to the padeye where the long link had broken, but he was unable to do so, because of the motion of the pontoon and poor visibility. This diver reported that the pontoon was now about level, indicating that the white end had slipped off the deck of the *Squalus,* so that the pontoon was now resting entirely on the bottom.

The first diver on the 31st succeeded in attaching the 1-inch wire rope lifting line to the white end of the pontoon. The second diver opened the vent at the red end, but no air bubbles came up. The red-end compartment was completely flooded. As this diver was being brought up, a rain squall hit the salvage area, the wind reaching a velocity of 58 knots for a short while and building up such a rough sea that operations had to be suspended for the day.

Early on the 1st of August, an attempt was made to lift the pontoon, the red end was raised about 10 feet, but the combined power of the forward capstan and anchor windlass was unable to start the white end, indicating that the center compartment was also flooded. Seven dives were made that day for the purpose of attaching an air hose to the center compartment blow valve. The first three divers were unable to connect up the hose due to battered threads on the pontoon fitting. The next three, working with new fittings, were also unsuccessful because the two halves of the union were made by different manufacturers and were not interchangeable. The seventh diver took down a new union that had been mated on the surface and succeeded in making up everything except the union, when, due to an increasing sea and approaching darkness, diving operations were suspended for the day.

The second diver on the following day, 2 August, made up the hose connection to the center compartment blow valve, but was unable to open the valve. The third diver discovered that the valve

was already open, which explained the flooding of the center compartment, because the hose to this connection had carried away during the lift of 13 July 1939. The fourth diver removed a pipe plug from the center compartment tell-tale fitting and opened the valve. The tell-tale pipe extends down to the bottom of the center compartment, thus providing a means of unwatering it by putting an air pressure on the center compartment. After blowing air into the center compartment via the blow valve for 45 minutes, air bubbles came to the surface, indicating that it was empty. Two hours later the pontoon was brought to the surface, the end compartments were then blown empty, and the pontoon was unrigged and started on its way to the Navy Yard in tow of the *Sagamore*. It had taken 8 days of work and 34 dives to raise this pontoon.

After the removal of the last pontoon from the stern of the *Squalus*, the next two days were spent trying to blow No. 4 main ballast tank. The hose originally attached to this tank had been pulled off by the small spare reeving line under the stern that had fouled it, thus stripping the threads of the air salvage fitting and so distorting it that it would not take the fitting on the lower end of the air hose. A special fitting had been devised by which an air hose was temporarily connected to the salvage connection to No. 4 main ballast tank and this tank blown down, after which the deck valve was closed, and the special fitting and air hose removed.

In the light of the experience gained from the lift of 13 July, it was decided to have more control pontoons at the upper level at both the bow and the stern to insure against a repetition of that disaster on the next lift. Accordingly, the arrangement shown in Figure 4 was adopted. This provided for three pontoons at the upper level at bow and stern and, in addition, furnished an ample margin of buoyancy for lifting either end of the *Squalus*.

This time all three slings were rove under each the bow and stern at one time to avoid the previous trouble of a fouled reeving line. One of the tail ends of each stern sling was attached to one side of a triangular-shaped reeving plate, to the opposite corner of which the down-going end of the reeving line was secured, and on the 5th of August the three sling assemblies were hauled under the stern together. The work of setting the six pontoons at the

stern was completed on the 8th. Bad weather interfered with work on the 9th, but on the 10th the three bow slings were rove under. The work of setting the bow pontoons was carried on day and night to take advantage of the fair weather, the work of setting the last pontoon being completed at 3:55 A. M. on the morning of the 11th.

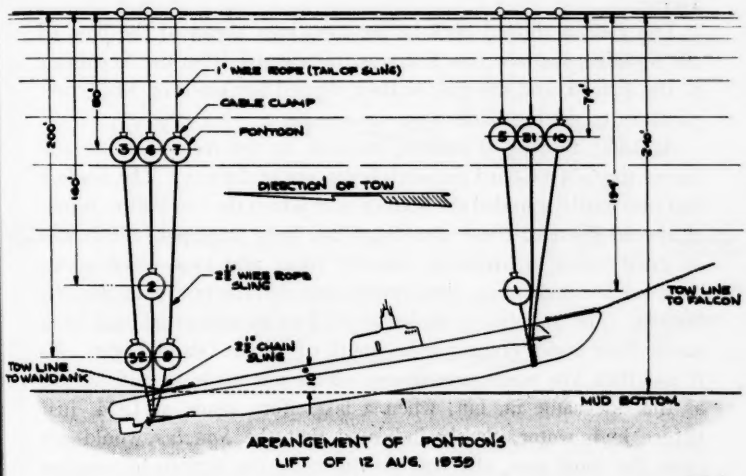


FIG. 4

The next day, the *Falcon* took position off to one side of the *Squalus* for the lift and the additional lengths of hoses required during the tow were attached to all air hoses to the submarine and pontoons and the other ends connected to the air manifold on the *Falcon*. Preliminary blowing started at 2113 and at 2341 was complete, all the upper pontoons, except one at the bow, having been blown dry.

The weather was ideal on the morning of 12 August. The final blow was started at 0628, beginning with the last unblown pontoon at the bow, after which No. 3 main ballast tank was blown dry, then the lower pontoons at the stern until at 0946, while blowing the last one, No. 32, the stern lifted, the upper pontoons coming slowly to

the surface, then submerging again for a short time in the white water caused by escaping air, and finally reappearing and riding in an awash condition. Additional tanks were blown down to make the upper pontoons at the stern ride at about half buoyancy.

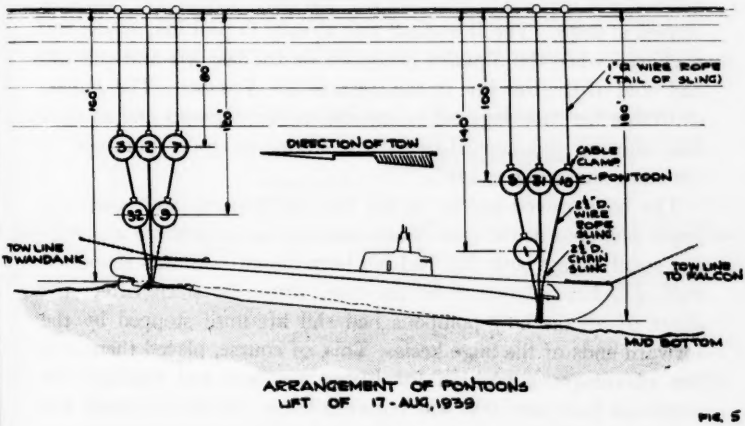
The lower pontoon at the bow was blown, followed by No. 1 fuel oil tank. When the latter was about blown, the bow lifted at 1151, surfacing the three upper pontoons, which soon rode well awash.

The *Falcon* shifted back in her moorings ahead of the bow of the *Squalus,* the bow tow line was transferred from the *Sagamore* to the *Falcon,* and the *Falcon* then slipped her mooring lines, preparatory to getting underway.

At 1300, the tug *Wandank,* assisted by the tug *Penacook* and one of the Coast Guard despatch boats, got underway. The *Sculpin* had previously sounded the bottom over which the tow was to move. A special chart of these soundings had been prepared, which was of great value in avoiding pinnacle rocks and local shoal areas. As a further precaution, the *Sculpin* preceded the tow, sounding the bottom. The *Squalus* grounded at 1413 on an uncharted mud bank about three and a quarter miles south of White Island Light. As it was then low water, continuous effort was made to pull her off as the tide came in, but, when a last effort, made at 1924, just before high water, clearly indicated that the *Squalus* would not clear the shoal spot, she was set down on the bottom by venting back and reflooding all pontoons and most of the ballast tanks. The stern pontoons remained on the surface, but the bow pontoons were pulled down about 20 feet below the surface, showing that the *Squalus* had grounded on a sloping part of the bottom. During this time the wind had risen to 23 knots, but, by skillful seamanship and the assistance of tugs, the *Falcon* was kept clear of the pontoons until new moorings were laid by the lighthouse tender *Hibiscus* and the *Sagamore.*

The wind continued throughout the night and during the next day, making the sea too rough for any work other than untangling hoses and keeping them clear of the pontoons. During the afternoon, the Salvage Area was visted by the President of the United States on the U. S. S. *Tuscaloosa,* but he was prevented by the bad weather from boarding the *Falcon.*

Since the bow was about 20 feet lower than the stern, it was now necessary to reset the upper pontoons at the bow to a depth of 100 feet, instead of 80 feet as planned, the arrangement for the next lift being as shown in Figure 5. This was done on the 14th of August, eighteen dives being necessary.



ARRANGEMENT OF PONTOONS
LIFT OF 17-AUG, 1939

FIG. 5

The next day, the two outside upper pontoons at the stern were lowered and reset alongside No. 2, already at the right depth on the middle sling at the stern, and the middle upper pontoon, No. 6, unrigged and sent back to the Navy Yard. Due to the angle of the pull now exerted by the slings of the two outside pontoons at the new upper level at the stern, the middle pontoon, No. 2, was forced down at one end. It took all of the 16th to correct this condition, it being finally accomplished by blowing the middle pontoon to full buoyancy to hold it up into position between the other two. In the meantime, two damaged air hoses to the submarine had been replaced. That afternoon the increasing southeasterly wind had raised such a choppy sea that operations had to be suspended. A thunder storm came up during the night, attended by a wind of 45 knots, which badly entangled the air hoses.

The morning of August 17th dawned clear, with very little wind. Strenuous efforts were made and all hoses untangled, checked, and

found intact with the exception of one, which had a surface leak that was quickly repaired. While this was going on, one of the bow pontoons that had upended was levelled off by pulling up on the white-end sling with the capstan, assisted by blowing air into the white end.

The weather continued propitious when this work was completed, hence it was decided to make the lift that day. Preliminary blowing started at 0928. The *Wandank* was already in position, as she had anchored when the *Squalus* grounded on the last lift and she still had hold of the tow line to the stern of the *Squalus*. The *Falcon* shifted in her moorings off to one side of the *Squalus* and at 1155 final blowing was started to lift the stern, the upper pontoons at the stern surfacing at 1407.

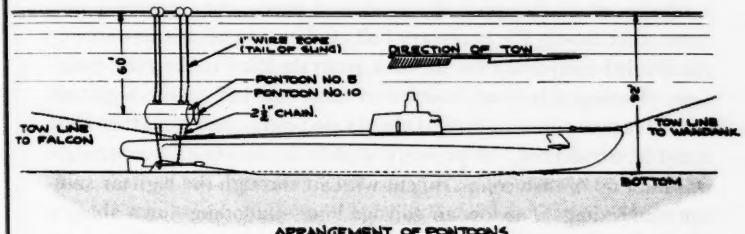
The bow proved harder to lift than anticipated and when the upper pontoons at the bow finally surfaced at 1710 there was only one small ballast tank that had not been blown. As was suspected then, and later shown to be the case, when the stern lifted, the slings of all the bow pontoons had slid aft until stopped by the forward ends of the bilge keels. This, of course, placed them at a less advantageous position for lifting the bow and explains the additional buoyancy that was required there. It also accounts for the apparent lightness of the stern, which had made it necessary to reflood No. 3 main ballast tank partially while blowing was in progress to raise the bow, and explains the damage sustained by the forward part of the conning tower fairwater, which was bashed in by pontoon No. 1 at this time.

The tow was started at 1740. At 2120 the *Squalus* grounded in 92 feet of water, one and six-tenths miles bearing 282 degrees true from White Island Light, in almost the exact location previously picked out. She was again set down firmly on the bottom, which later proved to be hard sand and level, although beset here and there with large boulders. The *Hibiscus* and *Sagamore* again laid down moorings for the *Falcon,* to which she was secured by 2315.

The *Squalus* was now in sufficiently shallow water to permit diving on air and the use of two divers at a time for relatively long periods on the bottom. As most of the air hoses had been attached to the *Squalus* all Summer and many were known to be badly chafed by the action of the sea and, as there were a large number of

diver's descending lines, spare reeving wires, etc., entangling the topside of the *Squalus,* it was decided to renew all air hoses and clear her topside of all entanglements.

There were several possible ways by which the final lift could be made. After careful consideration, in which the probability of increasingly bad weather played an important role, it was decided, as shown in Figure 6, to remove as much water as possible from all flooded compartments, rig two pontoons at the stern in a fore and aft direction and down as close to the deck as possible to furnish



ARRANGEMENT OF PONTOONS
LIFT OF 28-AUG, 1939

FIG. 6

what additional buoyancy was still required to lift the stern, and depend on the fuel and ballast tanks to lift the bow. It was known from the start that there was a good probability of this plan failing, but, if so, it was expected that only a couple of days' work during good weather would be required to rig any additional pontoons necessary.

Accordingly, the work of unrigging the bow pontoons was started on the 18th, but, due to dense fog and choppy seas, and the fact that now there was some 90 feet of 2½-inch wire rope and a like amount of 1-inch wire rope on each sling above the upper pontoons that had become thoroughly entangled with the pontoons during the previous lift and tow and which now greatly hampered the unrigging of pontoons, it was late on the 19th before the fourth, and last, pontoon at the bow had been raised and unrigged. It took three more days to unrig the stern pontoons. Here the work was further complicated by the fact that the lifting eyes in the wedges of both cable clamps of one pontoon had broken off, hence

the cable clamps could not be released from the sling. This pontoon was placed on negative buoyancy and lowered down a few feet by its lowering lines from the *Falcon,* one of the cable clamps being held up well clear of the pontoon by means of a strap around the body of the cable clamp. A diver then burned off the 2½-inch wire rope sling below this cable clamp by working from the top of the pontoon. This permitted unreeving the sling by hauling up on its intact end, thereby releasing the pontoon. As the last sling was hauled out from under the stern, a spare reeving wire was hauled under.

The work of stripping the topside of hoses, etc., attaching new hoses, and closing the necessary hull openings to permit unwatering the flooded compartments aft, took from the 23rd to the 26th, inclusive. During this time, hoses were also attached to the high and low salvage air lines to each of the flooded compartments aft so they could be unwatered. A pressure slightly in excess of sea pressure was built up in each compartment with air through the high air salvage connection. The low air salvage lines, extending down about a third of the way in each compartment, were connected by an air hose to a special manifold on the *Falcon,* from which the *Falcon's* wrecking pump could take suction. Unwatering of all flooded compartments was started late on the 26th of August and continued day and night until the next lift was made.

The two pontoons at the stern were set on the 27th, not, however, without considerable trouble in reeving the slings. The reeving line had taken a turn around one of the propellers and was foul of the chain of the sling which broke during the first attempted lift and which had apparently been caught under one of the pontoon slings on each of the other lifts and was thus carried along with the *Squalus.*

Late that day the *Falcon* took her position for blowing, about 150 feet to starboard of the *Squalus* and on the same heading, the plan being to tow the submarine bow first this time. Extra lengths had been attached to all salvage hoses by 2330. The preliminary blow, by which only three fuel oil tanks could be safely blown down without danger of prematurely lifting the submarine, was started at 1958 and completed at 0050 on the 28th.

The assisting vessels started taking position at 0615 on August 28th, but, due to the strong set of the tide, were not in position until 0945, eight minutes after which the final blow was started.

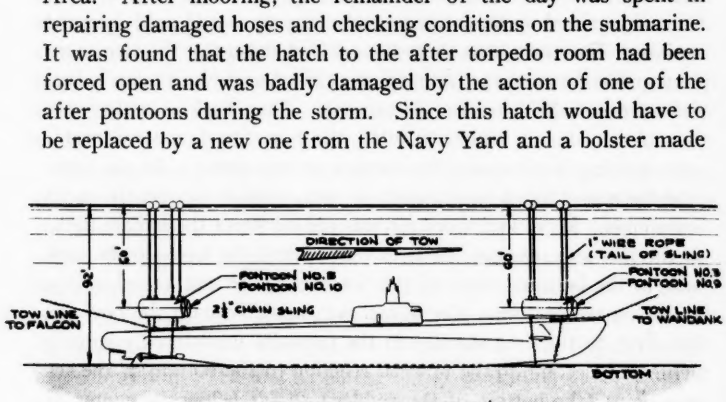
The plan was to lift the bow first with enough partially blown ballast tanks when it started up so that, by the time it surfaced, the buoyancy gained by these tanks on the way up would hold it on the surface. The stern was then to be lifted, bringing the two stern pontoons to the surface. Accordingly, No. 1 and No. 2 main ballast tanks were each blown for 15 minutes. Two fuel oil tanks were then blown, during which, at 1044, the bow lifted, a considerable part showing itself above the surface of the water. At the same time the vessel listed slowly over to port, until it was nearly on its beam ends. More and more air was spilled from the ballast tanks on the low side, causing them to reflood and the bow slowly sank back to the bottom, some of the list coming off the vessel as she settled on the bottom. The vessel was unstable with the bow up, due, first, to the free surface in the partially unwatered compartments and, second, to the upward reaction of the bottom of the sea on a point (the bottom of the rudder) so far below the center of gravity.

It was hoped that, by lifting the stern first, the two pontoons at the stern would right the submarine, after which the bow might be raised, during which the two pontoons at the stern would keep the vessel upright. At 1128, while blowing the stern pontoons, the stern lifted; the pontoons surfaced and took most of the list off the ship. But the free water in the partially unwatered compartments ran forward, making the bow too heavy to lift with the available buoyancy, hence at 1450 tanks were reluctantly vented back to sink the stern, which disappeared at 1518. The tank pressures indicated the *Squalus* had a 30-degree list to port, which was later found to be 34 degrees when measured by divers.

Decision was made at once to rig two pontoons at the bow in a fore and aft direction and as close down to the deck as possible, as indicated in Figure 7. One sling was to be placed forward of the bow planes on each side, the other abaft them on each side, the slings to be lashed to the submarine to prevent reeving around or slipping fore and aft.

Fresh northeasterly winds on the 29th prevented working on pontoons, but four dives were made to check up on conditions

below. During the night and next day the wind increased to such an extent that at 1246 the *Falcon* had to slip her moorings and seek shelter in the Navy Yard where she was held by weather until 0645 on 1 September, at which time she got underway for the Salvage Area. After mooring, the remainder of the day was spent in repairing damaged hoses and checking conditions on the submarine. It was found that the hatch to the after torpedo room had been forced open and was badly damaged by the action of one of the after pontoons during the storm. Since this hatch would have to be replaced by a new one from the Navy Yard and a bolster made



ARRANGEMENT OF PONTOONS
LIFT OF 13-SEPT, 1939

FIG. 7

up there to protect the new hatch, the time, while awaiting the arrival of this material, was utilized in starting a tunnel under the submarine to determine the feasibility of attaching more pontoons, should this be necessary. The bottom proved to be coarse gravel covered with sand and shells and such slow progress was made on the tunnel that after seven dives it was abandoned.

The new hatch cover for the after torpedo room was installed on the 3rd and the protecting bolster fitted in place abaft it. The next day, No. 4 main ballast tank was blown through one of its flood valves, a diver having accidentally twisted off the stem of the air salvage valve to this tank while closing it a few days previously so that the regular salvage air connection could not be used. The slings of the after pontoons were lashed to the port propeller shaft on the 4th to prevent their reeving should the submarine still have a list when the stern lifted, thus causing one pontoon to surface before the other.

An attempt to reset the after pontoons to their correct level and closer to the submarine had to be given up due to the rolling of the *Falcon* which nearly carried away a new set of pontoon lowering lines. The northwest wind increased during the night and the next day and at times reached a velocity of 40 knots which prevented all work, the *Falcon* having dragged one of her moorings. The wind went down during the following night but had increased to 20 knots at 1038 on the 7th when the *Falcon* started dragging her moorings again. As conditions grew worse, the *Falcon* again slipped her moorings and proceeded to the Navy Yard, but was able to get underway for the Salvage Area at 0645 next morning.

The after pontoons were reset to correct levels on the afternoon of the 8th and all air hoses again checked for leaks. On the 9th a bolster was set in place to protect the hatch to the forward torpedo room from the bow pontoons, one end of which would be directly over it. Lashings for the bow pontoon slings were also attached to cleats and kingposts on the submarine. The slings for the two bow pontoons were rove and the two bow pontoons set, a total of 26 dives having been made that day, the operations continuing until 2218.

During the 10th the bow pontoons were leveled off and the lashings, previously attached to the submarine, were shackled to the slings. Finally, the bow buoyancy tank was blown down by sticking an air hose up into one of the flood openings at the bottom of the tank. Again operations were cut short by a strong wind which reached a velocity of 38 knots.

The wind finally moderated to 18 knots by 1630 on the 11th, at which time a pair of divers renewed a damaged air hose to the after torpedo room. As this virtually completed the work necessary for the final lift, blowing and pumping of the flooded compartments was started at 2030 and continued until the final lift was made. The next morning was spent in correcting leaky air connections on the submarine. At 1230 the *Falcon* moved aft in her moorings astern of the *Squalus*, preparatory for the final lift. The extra lengths of hoses, required during the tow, were attached during the late afternoon and night. The preliminary blowing was started at 2235 and finished at 0148 on the 13th.

This time the stern was to be lifted first, sufficiently to surface the two stern pontoons, then the bow was to be lifted until the two bow pontoons surfaced, after which the *Squalus* was to be towed to the Navy Yard, the blowing procedure being timed to bring her into the Lower Harbor at high slack water as it was very doubtful whether the water would be deep enough on the way in except at high water.

The final blow to lift the stern began at 0548, starting with the two after pontoons and followed by the main ballast tanks and fuel oil tanks in succession from aft. The *Squalus* was gradually righted as more and more tanks were blown down and at 0802 the stern rose about half way to the surface, spilled air from her ballast tanks, and resank. After the spilled air had been replaced and while check-blowing the after pontoons, the stern lifted at 0915, surfacing the stern pontoons. The center compartment tell-tale valve on the port after pontoon had come open on the way up and was spouting air and water. This valve was soon closed and the pontoon flood valves closed.

Blowing to raise the bow was started on a pair of No. 2 main ballast tanks and the bow pontoons. At 0955 the bow lifted. The forward pontoons, the bow itself, and a part of the conning tower fairwater surfaced, but, due to the lightness of the bow, the bow pontoons were unable to exert any righting effect on the submarine, and she gradually heeled over to starboard, at the same time spilling air from all blown main ballast tanks, until finally at 1005 the stern pulled the after pontoons under and sank back to the bottom despite all efforts to keep her up and right her by blowing starboard tanks and venting port tanks.

An effort was made to raise the stern again but failed even though all tanks were blown down as far as the list of the ship permitted.

There was nothing left but set the bow down on the bottom again and start over, taking care this time to minimize the number of partially blown tanks while blowing was in progress to raise the bow. At 1030 the flooding back of main ballast tanks and pontoons was started, the bow sinking to the bottom at 1209.

Blowing was at once started to lift the stern, using all the fuel tanks and No. 3 main ballast tank. As blowing continued, the list

was again gradually reduced, but, with all the after tanks and the stern pontoons blown, the stern failed to lift. The flooded compartments had obviously reflooded somewhat during the previous lift. Failure seemed inevitable, when, after carefully topping off all tanks, the stern slowly lifted, the stern pontoons broke surface and disappeared only to reappear shortly thereafter barely awash. Further topping off made possible by the decreasing list, and blowing of tanks forward to raise the bow gradually increased the margin of safety at the stern. The bow pontoons surfaced at 1505 and the tow got underway at 1512. High slack water had long since come and gone, hence speed was adjusted to bring the tow into the harbor just before low slack water which occurred at 2000.

The *Squalus* grounded three times on the way in, her draft of nearly 40 feet being slightly greater than the available depth in the channel at these places. Each time the navy tug *Wandank* and the privately owned tug *Chandler* succeeded in hauling the *Squalus* off these shoal spots and, by skillful handling of the tow, the *Squalus* was grounded about 100 feet off Berth 6 at the Navy Yard about 1940. As the tide came in, she was warped alongside berth six with her conning tower abreast the 100-ton shear legs crane.

A hole was burned through the fairwater plating of the conning tower and a heavy wire rope strap passed through the periscope supports on which a strain was taken by the hook of the shear legs to hold the submarine upright while she was being surfaced.

The bow was then made buoyant by blowing No. 1 main ballast tank and at 2355 it slowly emerged, thereby unloading the slings of the two bow pontoons, which were unrigged and hauled clear. As soon as the hatch to the forward torpedo room was well out of the water, it was opened and air-driven submersible pumps taken below. Working from the forward torpedo room aft, the hull compartments were unwatered in succession back to the after battery room. By this time the bow was well out of the water, but as the tide dropped, the stern grounded, thereby reducing the trim and at the same time reducing the stability to such an extent that it was impossible to keep the *Squalus* upright without danger of overloading the shear legs, and the *Squalus* took a big list. In the meantime, burners had cleared away the wreckage over the conning tower hatch, which was opened up to permit the introduction of suction

hoses from the wrecking pump on the tug *Penacook* and the Yard fire engine, the air-driven submersible pumps being kept busy also. The water level in the After Battery Room was well above the door to the Control Room, but this door was forced ajar by jacks, thus permitting the water to run into the Control Room, from which it was pumped overboard.

The water level in the After Battery Room was soon lowered enough to permit opening the door into this space, after which suction was taken direct from the After Battery Room. The water level there was slowly lowered, but it was soon evident that water was entering the vessel nearly as fast as it was being pumped out.

Meanwhile, the tide was again rising and the bow was again up out of the water at a fearful angle, but the stern was still so heavy that the two stern pontoons were pulled half under.

Officer volunteers from the Navy Yard stripped and swam back through the oily water to the after bulkhead of the After Battery Room where they discovered that water was flowing into this space from aft through the hull ventilation supply and exhaust ducts. Several attempts were made by them to close the bulkhead valves in these ducts, but they were hopelessly jammed and could not be budged. The engine exhaust valves were suspected of leaking. One of the Yard divers attempted to gag them closed, but, due to various reasons, he only succeeded in closing and gagging the two after ones.

It was obvious that the source of leakage into the after part of the vessel would have to be found and eliminated, else the stern could not be raised. The hoses and portable lights were accordingly removed from the After Battery Room and the watertight door between it and the Control Room closed to permit putting an air pressure on the after compartments. It was now late in the afternoon of the 14th and the tide was again low, causing the *Squalus* to heel over about 30 degrees to port as the stern again grounded.

Air pressure was put on all the hull compartments abaft the Control Room, which soon disclosed that both engine exhaust valves from the Forward Engine Room were leaking. The starboard one was now out of water due to the bad list on the ship and the grounding of the stern. This valve was soon closed and gagged,

after which air pressure was again put on the after compartments. The outlet of the port engine exhaust valve inside the boat was so low down due to the port list, that a large quantity of water was forced out of the boat through this valve as well as the after torpedo tubes. It was soon evident that the stern was rising and about 2203 the entire deck was in sight. This blowing was continued until the *Squalus* was well afloat. All the hatches in the after part of the vessel were then opened to permit pumping direct from each flooded space and at 0240 on 15 September, the *Squalus* entered the dry dock.

The salvage was over except for reconditioning. Practically everything, except delicate instruments and the electrical installation, was in a remarkably good state of preservation. The damage incidental to salvage was insignificant and quickly repaired. The reinstallation of equipment is well underway and in a few months the *Squalus* should be back in commission as the U. S. S. *Sailfish*.

In conclusion it should be mentioned that, in spite of the hazards attending the 640 dives in depths down to 240 feet and the handling of such awkward and heavy weights as pontoons, pontoon slings, heavy bolsters, etc., there was not a single injury which incapacitated a man for more than a few days. There were only two slight cases of bends, in both of which the diver was able to go back to duty the following day. Needless to say, this was a source of great satisfaction to all those concerned in the work.

The author is extremely indebted to Commander A. I. McKee, (CC), U. S. Navy, for his kind suggestions and criticisms of this paper during its preparation.

ELECTRICAL INSULATION.

By Jesse B. Lunsford, Member.*

This article is the second in a series of four which question the older conceptions of electrical rating and propose a reduction in the weight of shipboard electrical machinery. The author develops the importance of insulating varnish in its relation to electrical, physical, and chemical surroundings. He recognizes that each of these environments demands ability characteristics which differ from and at times combat each other. A compromise does exist; individual values are sacrificed for better balance. But the author supports a conviction that more carefully directed sacrifice will result in a material of improved compromise and thus contribute to the ultimate aim of greater power for less weight.

PART II—INSULATING VARNISH.

THE ARCH.

In principle, the arch is old, important, respected; the Greeks had a word for it. Even as a prefix it carried a punch, making a *chief* out of anything with which it was associated; irrespective of whether that something was *good* (archangel, archbishop) or *bad* (archvillain, arch enemy) or just plain *old* (archaic). And, as every one familiar with the principles of electrical design will agree, the service reliability of electrical machinery is *chiefly* a matter of being only as *good* or *bad* or *old* as its *insulation*. That makes *insulation* an *archsomething* of electrical material.

And it likewise makes electrical insulation both the *archvillain* and the *archangel* of this piece; because it is both the chief obstacle and the chief hope of putting two horsepower where only one was before. The reason is simple; insulation is the chief (in fact only) support of electrical conductors.

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But the above references to *arch* were not meant to be a mere play on words, as a closer examination of the general principles will reveal. Rather the arch typifies the best possible utilization of a structurally weak material to support a heavy load; and the analogy is thus to the role which insulation, relatively weak in its mechanical properties, must play in any assemblage of far stronger materials such as copper and steel. Every pound of force or torque developed by virtue of the interlinkage of the electric (copper) and the magnetic (steel) circuits must be *transmitted through the insulation.*

Thus, to the dielectric stresses of insulation, there must be added the physical stresses of the far stronger metals (copper and steel). Already the weakest of the trio, it must carry a triple burden (dielectric, thermal, mechanical). The insulation that lies in the armature slots, between the copper and the steel, must stand up under the same pounding as the teeth of the gears in the same electro-mechanical assembly.

The arch, as a principle of mechanics, had its beginnings in stone; where, in the bridge or viaduct or building, it supported a transportation system or housing. What else does insulation do but support, at the required elevation (potential), a transport or housing system for electricity? And the main trouble is that insulation, at its best, is so *stone-like* in its weakness against impact that it is and must continue, in most applications, to be used in compression, as is stone.

If one doubts the applicability of these *stone-like* analogies to insulation, let him ponder well certain further facts. First that the very best inorganic insulators, from the dielectric angle, such as quartz, mica, glass, porcelain, asbestos, etc., are just stone, whether one looks at them in terms of their chemical composition or their tendency to shatter under impact. In fact they are mostly classifiable as silicates, and what else are stones?

The other class of dielectrics, the organics, are likewise at their electrical best when they are in the hard and stone-like condition. Every effort to soften or toughen them mechanically, as by the use of plasticizing agents, is at the direct expense of the electrical properties. And even so, as the plasticized organics are subjected to the heat, the physical and the electrical stresses in service, and

Tabulation of Electrical Insulating Rocks or Stones.

INORGANIC ROCKS.	ORGANIC ROCKS.
Asbestos	Amber
Aluminum oxide	Asphalt
Calcite	*Bakelite
Chalk	Beeswax
Concrete (Portland Cement)	Celluloid
Clay, bentonite (colloidal) (*alsifilm)	Cements (organic)
Feldspars	Copal
Glass (including glass fibers)	Oil, linseed (oxidized and polymerized)
Granite	Oil, tung (oxidized and polymerized)
Gypsum (plaster of Paris)	Paraffin
*Isolantite	Pitch
*Lava	Rosin
Marble	Rubber (hard)
Mica	Shellac
*Mycalax (glass-bound mica)	Synthetic Gums
Porcelain	Synthetic Resins
Quartz	Synthetic Waxes
Slate	Treated Papers, Fabrics and Fibres (wood, cotton, silk, etc.)
Soapstone	
Sodium Silicate	
Steatite (magnesium silicate)	
Sulphur	
Talc	
*Transite (asbestos-cement)	

* Trade names included merely because they are in such common use as to be classed almost as generic. They add nothing, however, to the tabulation, because the three inorganic ones (Isolantite, Mycalax and Lava) listed are simply man-made combinations of two or more of the silicates already listed separately. If the numerous other well-known combinations were listed, it would merely make the list longer without adding anything to its completeness.

by long exposure to the air, they eventually oxidize or lose their plasticizer. The length of time at a given temperature that it takes even the organics to become as brittle as stone is known as their life.

So, even those organic insulations which are not stone-like to begin with, must end up just that way. The thing that happens to the insulation that lies in the armature slots, between the copper and the steel, is that it finally shatters and blows away like just so much stone dust. So, again may it be pointed out that all these homely analogies to *stones* and *arches* are not mere figures of speech or a play on words. They epitomize, as well, certain basic conceptions which, to the author, represent the elements of a *philosophy of insulation;* at least as it applies to solid insulation.

A glance at the tabulation, wherein are listed the *foundation stones* on which solid electrical insulation rests, may reveal the source of that philosophy. That tabulation is basic, not superficial; and it is inclusive of practically all solid insulation, not just a few examples. If the reader doubts this, let him try to point out any stones that may be missing.

INTERESTING FACTS ON ROCKS.

1. Two metals, silicon and aluminum, together with oxygen (according to Bulletin 619, "Data on Geochemistry" of the U. S. Geological Survey, 1916) make up 82.92 per cent of the "Mantle Rock" comprising the earth's crust. Add two more metals (calcium and magnesium) to this combination and you account for 88.63 per cent of the earth's crust (mantle rock).

2. Most of that mass (88.63 per cent) is excellent electrical insulation, and practically all of it is satisfactory as insulation if you can keep water (as water) out of cracks of the rocks or off the surface of it. The water of crystallization does no harm.

3. The best insulators and the ones provided by nature in the greatest abundance are silica and the silicates. They are scattered about us in profusion, but we still do not know how to make the best use of them.

4. Our principal difficulty in such use is that these inorganic stones *refuse* to *bend* when we want them to. So we use the

organic stones, of which we have only a paltry few, and which we might estimate from some millionth of a per cent of the earth's crust. We seem to go out of our way for the much poorer organic stone when there is so much of the superior (electrically) earth rock at our feet.

5. Quartz, mica, asbestos, marble, etc., are the best inorganic electrical insulators. They are explained by the geologist in terms of heat and pressure and time, and classed as *metamorphic* rock. The best organic rocks are likewise formed by heat and pressure and time, and this molecular change is referred to by the chemist as *polymerization* or *condensation*. Call it *nature* when this metamorphosis happens underground, and *synthesis* when it is brought about in the laboratory, but please do not fail to think about it. The author said *stone* and means it.

6. Also remember that the first insulation uses of the organic materials listed above were not electrical; even primitive peoples learned that asphalt, pitch, beeswax, rubber (latex), etc., served to keep water off or out.

THE KEYSTONE.

In every arch there is one stone which is the key to its effectiveness. Such a stone is commonly referred to as the *keystone*, in recognition of its strategic importance to the whole structure. And it matters not how good the rest of the stones in the arch may be, if the keystone itself is crushed or shattered by the impacts of the loads which the arch carries. Hence this, the second in a series of articles on insulation, will take up that *keystone* as the first of the *stones* to be examined in detail, in the plan to promote a better arch; one strong enough to hold up two horsepower instead of one.

So few are they who have deigned to notice this homely member of the insulation family, this *gem stone* of the retiring disposition, this filler-upper of cracks and crevices, that the author takes great pleasure in thus announcing as his Cinderella selection—that sticky, slowly drying *stone* so often ignored:

INSULATING VARNISH.—For insulating varnish is just that—an *organic stone*, carried in solution, into the interstices of the other insulating stones, and there deposited and caused to harden into a

ASPHALT.

Courtesy Westinghouse Electric and Manufacturing Co.

RALEIGH CALKED SHIP AT TRINIDAD.

Asphalt now used for electrical insulation was just calking material for Sir Walter Raleigh's ships in 1617. Raleigh, sick and broken after an unsuccessful South American expedition, was on his way back to England and execution. He found the asphalt from the pitch lake at Trinidad good calking material for the seams of his ships.

The same asphalt lake has supplied tons of insulation for the world's electrical machines—and many miles of road surface. Some asphalt for electrical insulation now comes from mined gilsonite, and some from oil refineries.

solid *organic glass;* with just enough hardness not to crush or flow, and just enough softness (toughness, resilience, etc.) not to shatter under impact; and, to top it off, with just as much life as man's skill is able to put into it. It is an organic, *metamorphic* rock, that can be made to do some *bending.*

Now the most remarkable feature of all is that no matter how many other kinds of insulating stones there are in the windings of electrical machinery, and no matter how high their respective qualities may be, the only absolutely indispensable stone of any insulation assembly is this same insulating varnish. Various other insulating materials receive the star *billings* and *take the bow,* but it is the lowly insulating varnish which, in the final analysis, determines whether or not there is to be any bow for the star to acknowledge.

ESTIMATE OF THE SITUATION.

Since, then, in our battle for better insulation, we have to face a most worthy antagonist, in a strongly intrenched position, the situation calls for a careful estimate, an evaluation of the strengths, weaknesses and possible directions of its forces, to the end that we may expend our own efforts in the directions wherein it appears most probable (from experimental studies) that we can secure the maximum gains in the properties we must secure, with a minimum of sacrifice in any of those properties we can utilize. This being the case, let us proceed to look into this matter of insulating varnish from a number of different angles.

First, and in order to be as little biased as possible, in a purely objective study, let us assume that we know nothing whatever about the subject and hence must start from scratch by asking questions. For example, might we not inquire:

(*a*) What is an insulating varnish?
(*b*) What started the use of insulating varnish?
(*c*) Why is varnish now used?
(*d*) How does varnish compare with other insulations?
(*e*) How does varnish perform its services?
(*f*) How are insulating varnishes classified?
(*g*) How may we recognize a good varnish?

(*h*) As to purchase economy:

1. What is the best solvent to use?
2. What of the proportion of solvent to base?
3. What price longevity?
4. What is the *best buy?*

(*i*) As to the manner of its application:

1. What is the significance of *flash point?*
2. What is the significance of *specific gravity?*
3. What is the significance of *viscosity?*
4. What is the significance of *draining?*
5. What about *chemical* reaction?
6. What of *drying time?*

(*j*) As to the real services rendered by insulating varnish, what is the *significance* of the following characteristics:

1. Penetration?
2. Depth of drying?
3. Binding and adhesive properties?
4. High softening temperatures?
5. Chemical stability, or permanence?
6. Dielectric strength?
7. Power factor and dielectric constant?
8. Insulation resistance?
9. Water-proofness (fresh water, and moist air)?
10. Sea-water proofness?
11. Acid proofness?
12. Alkali proofness?
13. Oil proofness?
14. Life?

Let the reader not be misled by the length of this list into thinking that the author is going to attempt to answer all the above questions. On the contrary, every one of them is good for at least a dozen other such questions and half a hundred first-rate controversies. All the author seeks to do is to help take the subject of insulating varnish apart, to see what makes it tick and to learn if the varnish experts can put it back together again.

AMBER.

Courtesy Westinghouse Electric and Manufacturing Co.

Yellow Gum From Blue Earth.

King Tut's sarcophagus was varnished with melted amber. That was the hardest resin known then, and it still is. The ancient Phoenicians valued amber highly and braved the storms of the Atlantic in their triremes to gather it from the shores of the Baltic.

The Baltic is still the greatest amber region of the world. On the shores of East Prussia it occurs in colored sand called "blue earth." Amber was one of the first resins used for insulating varnish, but because of its high cost it has been replaced by other materials.

WHAT IS AN INSULATING VARNISH?

One good way to get the proper slant on the meaning of a word or term is to look in the dictionary. It appears, however, that this is one case where Mr. Webster is not of much use. The subject is somewhat complicated by the fact that the people who write dictionaries do not know anything important about insulating varnish, and the people who do know about varnish are unable to define it to the satisfaction even of their own colleagues or to get it put into any dictionary. In fact, any *definition* by one expert is a *"fighting word"* to the next expert in the same field.

The author has, however, managed to sneak a set of definitions from one expert, and to that extent he admits *plagiarism;* but he believes that he is safe from any such charges, because the expert who really formulated these definitions would probably be glad to escape the credit for them, and the other experts would almost certainly challenge their validity. So, without pretending that they are any creation of his own, the author gladly quotes them as something which, in the absence of something better, he subscribes to and willingly accepts all blame for.

GENERAL—INSULATING VARNISH.—An insulating varnish is a solution of drying oils, natural resins or gums, synthetic resins, asphalts, pitches, or nitro-cellulose, used singly or in combination in a volatile solvent. It may or may not contain driers, plasticizers or flexibilizers, and coloring agents, but does not contain pigments. When spread in a thin layer and exposed to air at room or elevated temperatures, it dries to a smooth, firm film by *evaporation* of the solvent; *oxidation, condensation,* or *polymerization* of the base; or by a *combination* of some or all of these.

DRYING-OIL INSULATING VARNISH.—An insulating varnish in which the principal ingredient is a vegetable oil, such as linseed or China-wood (tung oil), and which dries mainly by *oxidation* of the oil base following the evaporation of the solvent.

SYNTHETIC INSULATING VARNISH.—An insulating varnish in which the principal ingredient is a synthetic resin, such as the phenol-formaldehyde type or the alkyd type, and which dries mainly by *polymerization* of the resin base following evaporation of the solvent.

SPIRIT INSULATING VARNISH.—An insulating varnish in which the principal ingredients are natural gums or resins, such as shellac or copal, and which dries *mainly* by *evaporation* of the solvent.

INSULATING LACQUER.—An insulating varnish in which the principal ingredient is a synthetic lacquer resin base and which dries almost *entirely* by *evaporation* of the solvent.

WHAT STARTED THE USE OF INSULATING VARNISH?

A second and very good way of finding out about a subject is to study it from an historical angle, to look into its past. Fortunately for us, there was considerable explaining done in the early days when the little dynamos did not show up so well. The alibi was that the weather prevented them from living up to the specifications. Both the weather and the specification writers being unmanageable in those simple pioneer days, there was nothing left to do but to *use varnish*. The author, being a specification writer himself, gets a great kick out of such situations; how come it was not recommended that the specifications be changed to call for the tests to be made in dry weather?

Anyhow, the story runs about as follows. Early in the history of electrical machinery construction much trouble was experienced due to *insulation failures*. As a means of insuring himself against such failure the purchaser usually specified a high *insulation resistance*. The manufacturer found that although he could not keep it high permanently, he could make it high temporarily, for test purposes, by drying out the machinery just before the tests. As soon as this fact became generally known it was useless to depend, for serviceability, upon high insulation resistance requirements in a purchase specification. It was easy to get around such requirements, without any increase in the machinery's real serviceability, by simply drying it out a short time beforehand. The trouble was, however, that the benefits of drying out were extremely short lived.

Obviously, drying out machinery was not the real remedy for these insulation failures. It was resorted to at first merely as a temporary expediency; in some cases a subterfuge perhaps. Later it was accepted as a *clue*. After that, came some progress towards a *remedy*.

COPAL.

Courtesy Westinghouse Electric and Manufacturing Co.

Aged for a Million Years.

Vast tracts on the mainland coast of Zanzibar, once covered by primeval forests, now yield fossil copal or resin. Today not a single tree is visible along this coast, but at a depth of about four feet are valuable deposits of copal, formed from trees that lived a million years ago. Masses weighing as much as five tons are sometimes unearthed in one piece. Copal resin is used extensively in varnishes and electrical insulating compounds.

With the acceptance of moisture absorption as a clue to insulation difficulties, there came some experiments relating to the effects of such moisture. One investigator (Skinner, "Industrial Research," 1923) states the case very nicely as follows:

"In the writer's experience, the first appreciation came at an early date when it was discovered that measureable amounts of water could be distilled from apparently dry cotton covered wire. When the apparatus in which the wire was used was thoroughly oven dried and treated with varnish, insulation difficulties practically ceased. The appreciation of the part which moisture plays in insulation difficulties has increased with the years, and in some of the latest apparatus new researches have had to be undertaken which would provide ways and means of eliminating the most minute traces of moisture and provide against their reabsorption."

Other investigators (Turner and Hobart, "The Insulation of Electric Machines," 1905), in referring to this problem, point out "the enormous extent of the influence of moisture" and give curves showing the effect of drying on the insulation resistance and dielectric strength. In one curve it is shown that drying raised the insulation resistance of cotton duck from 4 megohms to 150 megohms. In another curve it is shown that the dielectric strength of Fuller Board was 10,800 volts "as received," that it was raised to 13,500 volts when "dried out," and still further raised to 22,000 volts when "varnished."

In a report rendered in 1905 by the Engineering Standards Committee of Great Britain, E. H. Raynor of the National Physical Laboratory reported the results of certain experiments with insulating materials. Among the data are certain bits of interesting information with respect to the "*Effect of Desiccation on Insulation Resistance.*" While this is no criterion of dielectric strength, it does show the enormous effect of extracting the small quantities of moisture that may be present in certain insulating materials. The materials investigated were Press-Spahn and Oiled Cloth. Several examples are given below:

(*a*) Press-Spahn, at atmospheric temperature: Insulation resistance 1.04 to 1.07 megohms in the "normal" condition, was raised to 6600 to 7000 megohms after 18 hours in the desiccator.

After 49 hours in the desiccator the resistance was raised to 41,000 megohms.

(*b*) Oiled Cloth, at atmospheric temperature: Insulation resistance 75 to 84 megohms in the "normal" condition, was raised to 1700 megohms after 2½ hours and still further increased to 48,000 megohms after 18 hours in the desiccator.

(*c*) Press-Spahn. Initial insulation resistance .42 megohms, rising to 3.4 megohms after 6½ hours, 510 megohms after 23 hours, 70,000 megohms after 76 hours, 200,000 megohms after 122 hours, and 800,000 megohms after 12 days in the desiccator.

(*d*) To show how quickly a desiccated piece of Press-Spahn reabsorbs moisture, a prepared sample gained in weight by 0.32 gramme after remaining three days in the "closed oven to cool," and 1.19 grammes after a total period of a week.

(*e*) To show how even exposure to "out-of-doors sunshine" helped as a desiccating agent for a sample kept indoors, a sample was tested before and after exposure. The installation resistance before exposure to the drying rays of the sun was 0.84 megohm, and after 70 minutes exposure it was increased to 630 megohms.

The quotations given above are purposely made from electrical literature which we now think of as out of date because so much progress has been made since it was published. One can resort to later papers for greater detail and a more refined technical treatment of the subject, but those few simple illustrations will show that getting the moisture out and keeping it out is one of the oldest problems, and yet also the spryest with which electrical men have to deal. Insulating varnish is the chief hope in this struggle.

WHY IS VARNISH NOW USED?

A third way of wrapping one's self around such an ill-defined subject as this is to find out why the people who now use insulating varnish think they should do so. Although such opinions are not necessarily a product of straight reasoning, the fact that sticking to such opinions is rather costly to them, in this case, and the further fact of the similarity of opinion expressed is a point worth considering. The reasons (or opinions) generally given for the continued use of insulating varnish are:

(*a*) To exclude moisture, oil, dirt, etc.
(*b*) To increase the dielectric strength of *composite insulation.*
(*c*) To increase the thermal conductivity of electrical coil windings.
(*d*) To act as a binding or bonding agent.
(*e*) To enhance the appearance and to promote the cleanliness of electrical windings.

The matter of how the above listed *results* are actually achieved by insulating varnish, what properties in a varnish are necessary to those ends, and how must varnish be applied, are all questions over which much controversy has raged. Since the author is not offering any answer at this point, the question of which of the above listed results is the more important, need not be raised now.

First, we shall study each of these in turn, with the idea of learning all we can about each.

The only matter of importance just at this point is to determine whether or not the five reasons (opinions, results) outlined above represent all there is to the question of "why is varnish now used?" The purpose of this question is to ascertain the scope of the problem so that the several details may be viewed in perspective. If there is either more or less to the picture of the problem than is outlined above, the author would appreciate learning of it; so he puts this question: "What else, if anything, is there to this subject?"

HOW DOES VARNISH COMPARE WITH OTHER INSULATING MATERIALS?

The question is sometimes asked: "Is varnish a better insulation than others; i.e., can it be used instead of or in the place of something else?" The author is usually loath to venture any direct, unqualified answer to any technical question, lest he live to regret it. In this case, however, he is willing to take a chance on one such as "no (not yet at any rate)," realizing that even that answer may properly be disputed. It will simplify the discussion, however, if that view be accepted for the time being, as the following will serve to illustrate:

Its physical limitations as a material do, at present, preclude the use of insulating varnish alone, in the majority of its applications. Furthermore, the only service, of the five listed above, which can be considered as purely electrical, is not usually the determining factor. It is evident, therefore, that its use has not been made imperative because of any purely electrical considerations. In a sense, therefore, its classification as an electrical "insulating" material has been a bit confusing.

The conditions which make its use imperative may be better understood if we will briefly review the circumstances under which varnish was first used and is even now universally used. Its first use was not to displace other dielectrics, and its present use is likewise not to displace but rather to supplement them. The inference is that other dielectrics are deficient in certain respects and that varnish is used to overcome, in some degree, those deficiencies.

An analysis of the usual designs of electrical machinery will show that the other dielectrics used (paper, cotton, silk, mica, asbestos, etc.) are provided in sufficient amounts to furnish a very high initial factor of safety for dielectric strength and insulation resistance. Some of them have an even higher dielectric strength than that of varnish, and yet varnish is used with all of them.

While the effective dielectric strength will apparently be increased by impregnation with varnish, this does not explain an imperative need because an increase in the dielectric strength of the machinery could be obtained in other ways; for example, by a change in the quality, or by increasing the amount of the other dielectrics used.

Even a great increase in the quality or the quantity of the other dielectrics used does not, however, solve the problem. Varnish would still be indicated just as imperatively as before the increase was made. Obviously, therefore, the explanation of the deficiencies is not to be found in a study of dielectric strength, per se. Other clues must be sought.

From this we may be justified in concluding that, for the present, at least, varnish is not properly to be compared with other dielectrics; that it is not in competition with anything else; and, therefore, our analytical efforts should be directed, rather, to the question of "How does it actually perform its required services?"

LINSEED.

Courtesy Westinghouse Electric and Manufacturing Co.

. . . It's Food in Abyssinia.

It's hard to believe . . . what we use principally as a base oil for paint and varnish, Abyssinians use for food. But linseed and linseed oil were used for food many centuries before they found their way into the paint and varnish business.

Boiled linseed oil in paints and varnishes acts as a vehicle to form the film. The oil itself is pressed from the seed of common flax, and heat is applied to increase the yield of oil.

HOW DOES VARNISH PERFORM ITS SERVICES?

Reverting, then, back to the five services stated hereinbefore as being claimed for varnish, let us examine each of these services in turn. It is to be assumed, of course, that not all five services are of equal importance; but all five are included in order that this estimate of the situation may not leave any stone unturned in getting at a true picture, a whole picture, and nothing but what should clearly show in the picture.

It is also obvious that the varnish cannot serve these five alleged services unless it is properly applied to the electrical windings it is to assist in the job of insulating. The methods used in these varnish applications are of such vital importance that this subject will be fully exposed hereinafter. There are only two factors in such application which require stressing at this point; i.e., the varnish *must:*

(*a*) have a clear right-of-way (road) in to its destination, and
(*b*) be forced or allowed to proceed completely to that destination.

At best it is not an easy matter for a viscous material like varnish to penetrate to the very innermost recesses of electrical windings. If the right-of-way in be blocked by air, moisture, oil, dirt, etc., the varnish simply can not get in; it is itself locked out by the very substance that it is designed primarily to exclude. No varnish can do any good without a fair chance. Let us, therefore, start out with the assumption that the varnish has been applied in such manner that it could do a good job if it were a good varnish. Thus we may concentrate on those factors which determine what a varnish must do to be good.

Claim (a). To Exclude Moisture, Oil, Dirt, Etc.

Now then, how could insulation varnish serve to exclude moisture, oil, acid, dirt, etc.? This should not be hard to guess; simply by playing the dog in the manger; just by:

(*a*) Getting there first.
(*b*) Occupying all the spaces.
(*c*) Doing no harm itself.
(*d*) Preventing harmful substances from entering.

Proper application permits it to *get there first* and in such quantities as to fill all interstices completely, so that there is positively no room for anything else. Procurement of the varnish under proper technical specifications will insure that it is capable of doing no harm itself. Thus the initial safeguards have been provided which prevent harmful substances from remaining in its rear or passing around it. The next step is to prevent them from going through.

Assuming that the varnish has been procured under proper technical specifications which insure the prevention of harmful substances from entering, these substances cannot *go through* as long as the varnish is of such a durable nature that it:

(*a*) Remains insoluble in and unaffected by moisture, oil, etc.
(*b*) Does not crack open under age, high temperature and physical violence.

The first is so obvious that it requires no further comment; the second does not, however, appear to be generally recognized.

Since varnish is composed of oxidizable organic substances, it begins to deteriorate as soon as the varnish film on the windings becomes dry. This process continues uninterruptedly until the life of the varnish is gone. The varnish is alive just as long as it is yielding and flexible. It is dead when it cracks open, and allows harmful substances to go through.

High temperature is a great accelerator of this decaying process. The temperature conditions in the machinery spaces of naval vessels are particularly severe. The machinery specifications for such vessels assume an ambient temperature of 50 decrees C. (122 degrees F.) and a maximum hot-spot operating temperature of 105 degrees C. (221 degrees F.) for some electrical machinery and 125 degrees C. (257 degrees F.) for other machinery. It can, therefore, readily be appreciated that the life of the varnish under these trying conditions is necessarily much shorter than in the usual, commercial, land installations.

Rotating armature windings are particularly susceptible to such injury, because there are not only constantly recurring expansions and contractions (corresponding to periods of operation and idleness) to be dealt with, but there also exist vibratory and centrifugal stresses which tend to *pulverize* the varnish and to sling it off like

so much *stone dust*. Any alternating current winding (rotating or stationary) is treated to considerable violence, depending upon the energy supplied to it. Armature windings in slots are comparable to gear teeth and transmit the same mechanical power in the same mechanical way. Materials that become embrittled with heat and with age have a hard life and a short one.

Dead varnish cracks open and allows the moisture to go through. Moisture loses no opportunity where cracks are concerned. Naval conditions make the cracks come quickly and also supply the moisture in plentiful amounts. It is a bad combination for the varnish. Only the highest quality of varnish has any chance of meeting it on anything like fair terms and an honorable conclusion.

Claim (b). To Increase Dielectric Strength.

Insulating varnish serves to increase the effective dielectric strength of the insulated electrical windings. The means by which this is accomplished are not thoroughly understood; but it appears that varnish acts to equalize (and, therefore, to lower) the unit potential stresses better in the composite insulation. Its specific inductive capacity (dielectric constant) is much nearer that of the other dielectrics with which it is used than is that of air or a vacuum. Its use to displace bubbles, layers or pockets of air, and to fill vacuous spaces, tends, therefore, to lower the potential gradients due to such inequalities; also the filling of the voids with varnish precludes ionization of the air and gases that would otherwise be contained in such voids.

Claim (c). To Increase Thermal Conductivity.

Insulating varnish serves to increase to a very marked degree the thermal conductivity of electrical windings insulated with the usual (fabric, paper, etc.) dielectrics. The thermal conductivity of air being only 1/10 to 1/20 that of the usual dielectrics or of varnish, the filling of all interstices in the windings with a compound having the higher thermal conductivity will lower the temperature gradient within the windings, and therefore permit them to dissipate more heat, or, for the same energy losses, effect a reduction in the maximum hot-spot temperatures. As heat is one of the arch enemies of insulation and the major factor in the rating of electrical machinery, any increase in heat transfer ability is of great importance.

TUNG (CHINA-WOOD) OIL.

Courtesy Westinghouse Electric and Manufacturing Co.

COOLIES ROAST SEEDS.

From the seeds of an oriental tree . . . comes one of the most important ingredients of insulating varnish, Chinese Tung Oil. Its importance can be measured by the fact that millions of pounds are sent to the United States every year. Chinese coolies gather the Tung seeds, roast them in flat dishes over open fires, and after coarsely powdering them, squeeze out the valuable oil in crude wooden presses.

Claim (d). To Act as a Binding Agent.

Varnish serves as a binding agent to tighten and bind the electrical windings into solid, impervious, unit-acting masses. The mechanical stresses are thus transmitted by the groups acting more nearly as groups, and without the chafing of adjacent insulated conductors upon each other, which happens where there is not such added mechanical interlocking. Electrical windings are usually either subjected to or transmit mechanical forces, and varnish helps to provide the necessary mass consciousness.

Claim (e). To Enhance Appearance.

Insulating varnish provides the smooth, hard coating over the electrical windings which so greatly enhances their appearance and also makes it harder for the dirt and dust to collect. Furthermore, such a coating makes the task of cleaning easier and more thorough when dirt does accumulate.

Summary as to Services Claimed.

It is clear from the foregoing that insulating varnish does perform the several services claimed for it. Some of these services are of so much more importance than are others that every effort should be made to avoid stressing the lesser services to a greater extent than is warranted, or the greater services to a lesser extent than they deserve. It may be of some help in this regard to think of these services as falling into two classes, viz., *routine, exceptional.*

Of the five services discussed above, all but one may be classed as *routine,* because they are readily performed by practically all insulating varnishes worth the name. It is a very easy matter to specify and to procure varnishes which are entirely satisfactory for these purposes.

The one service classified as *exceptional* is the "Exclusion of moisture, oil, etc." Only exceptionally good varnishes will serve this purpose under the severe conditions existing in the naval service. It is the paramount consideration from a naval viewpoint. It offers the greatest difficulty in the preparation of proper technical specifications and in the actual procurement of really satisfactory varnishes.

HOW ARE INSULATING VARNISHES CLASSIFIED?

Like almost every other present-day commodity, there are so many hundreds (even thousands) of different types, brands and compositions of varnish that one has to break these down into a few general groupings so as to avoid ambiguous or conflicting statements. For convenient reference, the several types of varnishes used in the construction and maintenance of electrical machinery may be classified according to the following general features:

(*a*) Color.
(*b*) Duty.
(*c*) Drying time.
(*d*) Manner of drying (hardening).
(*e*) Manner of application.

These classifications are merely to be thought of as convenient *handles,* to assist in the definition of certain varnish terms, and are not to be taken as indicating the suitability of any particular class or type of varnish. The question of what characteristics should be sought for a given purpose will be taken up later.

Classification (a). Color.

With respect to color, varnishes fall within the following classes:

Black.
Clear (transparent).

All light colored, transparent varnishes are classed as "clear."

Unfortunately, more stress is oftentimes laid upon the color of a varnish than this feature deserves. Some users have such a strong preference for a given color as practically to dismiss without thought all other technical considerations. One particular brand of varnish having a certain color may be better than another brand having a different color; but the color is merely incidental, it is no criterion of quality or of suitability.

Black varnish seems to appeal to the aesthetic sense of the majority of users, particularly in the naval service. There are probably two explanations: First, that it harmonizes better with the surroundings, and, second, that it has greater hiding power for dirt and inequalities. The first is difficult to evaluate, and the second is a distinct disadvantage from a technical standpoint.

From a maintenance viewpoint there are two reasons for preferring black if black varnish was used originally in impregnating the windings. One is that if a coat of clear varnish is applied over the original black one, the resulting color is not a particularly pleasing one. Another is that a repaired coil or section cannot be hid with the clear varnish, and so it remains looking exactly what it is, a repaired job.

Since appearance is not the main reason for the use of insulating varnish, it is neither logical nor desirable to subordinate all other considerations to it. Either color, or both, should be permitted, but the varnish, whatever its color, should be able to serve the purpose for which its use is imperatively demanded.

Classification (b). Duty.

GENERAL. With respect to the duty to be performed, varnishes fall within the following classes:

(1) Insulating.
(2) Finishing.
(3) Sticking.
(4) Core Plate.

INSULATING (1). The varnishes classed as *insulating* are those which are designed as the basic, impregnating compound primarily depended upon to exclude moisture, oil, dirt, etc., from the interior of the windings. It may or may not be necessary to apply "finishing" varnish over the "insulating" coat.

An *insulating* varnish is sometimes found sufficient for all purposes, but generally there is a need for both the insulating and the finishing types. The "insulating" type should always be used where there is a need for real protection. To meet all conditions and color preferences this type should be specified in both the "baking" and "air drying" and in both the "black" and "clear" types.

FINISHING (2). The varnishes classed as *finishing* are designed primarily as an additional coat to be applied over "insulating" varnishes. They have a smoother, harder, glossier film which is of a better appearance and more readily cleaned than that of "insulating" varnishes. Also, they are, in general and because of the harder film, highly resistant to oils, acids, etc. While, therefore,

the *finishing* type varnish may be used as an added protection, over the last application of an insulating varnish, it should not be used in lieu thereof. To meet all color preferences demanded in service the *finishing* type varnish may be specified in both the black and the clear colors.

STICKING AND CORE PLATE (3) AND (4). *Sticking* varnishes are designed for use in sticking down the ends of tape on taped coils to avoid sewing, and for sticking insulation to other parts as may be necessary. *Core plate* varnishes are designed for insulating the core laminations from each other, to prevent eddy current losses in the iron core. As these types of varnishes fulfill what has hereinbefore been classed as routine or incidental purposes, easily served, there is no need to enlarge further on them. If they could be made thrice as good as they now are, it would not help us much in our problem.

It will be noted that no attempt has been made to include in the foregoing such materials as shellac, spirit finishes, lacquers, solid impregnating compounds, etc. These materials are not properly classifiable as " insulating varnishes " although to a certain more or less limited extent they are used instead of insulating varnish for specific purposes.

CLASSIFICATION (C). DRYING TIME.

GENERAL. With respect to the *time* of drying, or difficulty of drying, varnishes are classified as follows:

(1) Air drying.
(2) Baking.

These classifications are somewhat arbitrary, because practically all air drying varnishes can be dried in a satisfactory manner and in a shorter time by baking, and all baking varnishes will air dry in time. A varnish is classified as " air drying " if the time of drying in air is short enough to render this method economically feasible, and as " baking " if the normal drying time in air is so long that high temperature must be used to accelerate the drying process so as to reduce this period to a reasonable figure.

As a rule, the varnishes which normally dry in a relatively short time have a comparatively short life (flexibility), and deteriorate

readily at high temperature. They are, however, of a more binding nature and have a harder (though brittle) film.

As a rule, the varnishes which normally require a relatively long time to dry in air have a comparatively long life (flexibility), and are capable of withstanding a higher temperature without detriment. Their films are somewhat softer and more resilient than the quicker drying varnishes.

The economical aspects of a given situation (facilities available, size and importance of part, character of service to be performed) will indicate if it is permissible to employ a quick drying (air drying) varnish, or whether it is necessary to use a slower drying (baking) varnish for the particular application.

BAKING VARNISHES. As a long drying time is usually associated with long life, and as there is great need of longevity under the severe naval conditions, it goes without saying that a baking type of varnish should be selected wherever it is practicable to make use of it.

AIR DRYING VARNISHES. In the case of much of the large electrical machinery which is already installed aboard ship, it may be impracticable to use a baking varnish. In such cases there is a real need for an air drying type of varnish which has characteristics as nearly like the baking varnish as it is practicable to get, and yet whose drying time is short enough to permit its satisfactory use under the usual conditions met with aboard ship. It should represent the best possible compromise between longevity in service and ease of service application. It also may represent somewhat of a compromise between the "insulating" and "finishing" types so that one application of varnish will serve both purposes for the usual maintenance work.

CLASSIFICATION (D). MANNER OF DRYING (HARDENING).

GENERAL. With respect to the method of drying (hardening), varnishes fall within the following classes:

(1) Those which dry through the simple evaporation of their solvent.

(2) Those which, after elimination of the solvent, dry by oxidation or harden as the result of some chemical change.

IMAGES WERE KISSED.

Courtesy Westinghouse Electric and Manufacturing Co.

. . . Mica Became "Insulation."

Mica was used for many peculiar purposes before it became an electrical insulator. One of the most interesting early applications as an "insulating material" was the glazing of Holy images.

Images of the Holy Virgin were much kissed by devout Christians in the 17th and 18th centuries. Wear and tear on the images were finally checked by glazing them with mica.

The first class are extremely rapid in their drying action. It is simply a case of deposition. The resulting coat has all the properties of the original gum contained in the solution.

The second class are relatively slow drying. Here it is a case of the transformation or conversion of liquid to a solid, as by condensation or polymerization.

Various compromises between the two classes listed account for the numerous degrees of binding power, hardness, high temperature endurance, life, etc. Gains in certain characteristics are made by sacrifices in certain others, depending upon the results desired.

In the case of varnish which dries through the simple evaporation of its solvent, the resultant film is hard, brittle, and readily affected by high temperature. Its flexibility life is practically nil. A common solvent is alcohol, which readily takes up moisture. If the solvent carries moisture with it into the innermost parts of the windings, or if the varnish has no flexibility life and cracks so as to let moisture in subsequently, it is obvious that it cannot possibly serve the main purpose for which varnish is used; i.e., it cannot exclude moisture, etc. Varnishes of this type are not suited to naval conditions and should not be selected for such service.

In the case of varnish which, after elimination of the solvent, dries by oxidation or hardens as the result of some chemical change, the resultant film is softer but much tougher and more resilient than is possible with that of the first class mentioned above. The heat endurance and flexibility life are relatively very great. It is able to bend and stretch, give and take, to meet the thermal expansions and contractions, and the physical punishment which electrical windings are usually called upon to endure. This type is susceptible of such modifications as will enable it to be utilized for both "baking" and "air drying" purposes. In view of its characteristics, this is the type which should furnish the groundwork for all navy insulating varnishes.

Classification (e). Manner of Application.

As the manner of application will, in general, determine the proper consistency for such use, varnishes may be still further classified as follows:

(1) For vacuum impregnation.
(2) For dipping.
(3) For spraying.
(4) For brushing.

For reasons of economy, varnishes are usually purchased on the basis of a stated consistency, this consistency being altered later to a value which experience has proved to be the most satisfactory for the work in hand.

The manner of application is such a large subject in itself, that it will be necessary to cover it in detail elsewhere and at another time. It has been discussed above only insofar as the manner of its application affects the type of varnish itself.

HOW MAY WE RECOGNIZE A GOOD VARNISH?

This is a good question, whatever one may think of some of the answers which may be suggested, because unless we have some very specific and constructive ideas as to what we are looking for in a varnish, how then would we be able to recognize it if we found it, or to improve upon it in any case. Thus it is obvious that we need not expect to find much unless we are prepared to contribute something in the way of intelligent questioning; an analytical approach to the subject. Fortunately, there are three simple angles to almost any such question, varnishes included, viz.:

(*a*) Purchase economy. (What is the best buy?)
(*b*) Manner of application. (How will we apply it?)
(*c*) Service rendered. (What do we get out of it?)

Purchase Economy.

It is easy to see that those technical considerations which directly affect purchase economy are:

(1) Nature of solvent.
(2) Proportion of solvent.
(3) Proportion of base.
(4) Life (heat endurance).

MANNER OF APPLICATION.

Likewise, those technical characteristics which directly affect the manner and safety of its application are:

(1) Flash point.
(2) Specific gravity.
(3) Viscosity.
(4) Draining.
(5) Chemical reaction.
(6) Drying time.

SERVICE RENDERED.

Similarly those technical characteristics which directly affect the nature and the amount of the service rendered by the applied varnish are:

(1) Penetration.
(2) Depth of drying.
(3) Binding and adhesive properties.
(4) High softening temperatures.
(5) Chemical stability or permanence.
(6) Dielectric strength.
(7) Power factor and dielectric constant.
(8) Insulation resistance.
(9) Water-proofness (fresh water and moist air).
(10) Sea water-proofness.
(11) Acid-proofness.
(12) Alkali-proofness.
(13) Oil-proofness.
(14) Life.

NOTES. (*a*) In cases where the finished electrical machinery is purchased from the machinery manufacturer, the Navy is interested only in those characteristics which directly affect the "service rendered." Those which affect "purchase economy" and "manner of application" do, however, become important when the Navy purchases varnish and applies it.

(*b*) It will also be noted that *life* has been listed both under "purchase economy" and under "service rendered" because it vitally affects both. With this one exception, however, a given

characteristic is only listed once, even though it affects more than one classification; in cases of dual effect the characteristic is listed according to its most direct effect.

(*c*) In the following notes, covering each of the characteristics listed above, the discussion will be confined to the general considerations, definitions and limitations which apply generally to such characteristics. Specific data on these particular characteristics will not be given at this time, because these are too voluminous to include in this article; they would be out of keeping with the strictly ideological purpose back of this discussion. These are *ideas,* not *details.*

As to Purchase Economy.

Reverting back to the first suboutline under the main question, it will be observed that a good buy must take four factors into consideration, viz., (1) nature of solvent, (2) proportion of solvent, (3) proportion of base, and (4) life. While it is almost impossible to separate the technical from the economy points of view, the following represents an attempt to picture both angles:

WHAT IS THE BEST SOLVENT TO USE? If the solvent is of such a nature that it will absorb moisture, this moisture will be carried into the windings which are being impregnated. If the windings which are to be varnished contain "enamelled" magnet wire, the solvent used may have such solvent powers as to soften the enamel insulation on such wire and cause partial short-circuiting of the windings. If the solvent used contains fractions which distill off at either very high or very low temperatures, difficulty may be experienced with the varnish after it has been repeatedly thinned in service; the filling power and drying time of the varnish will be affected. From a technical standpoint, therefore, these possible characteristics must be taken into consideration in determining the nature of the solvent to be used.

Assuming that proper precautions have been taken to insure suitable technical characteristics in the solvent, it is very important that the solvent be of such nature that it can be procured in accordance with high commercial standards, at a reasonable price and within a short delivery period. It makes considerable difference

whether the cost of the solvent is ten cents or a dollar per gallon, when lots of gallons are required. The solvent must be renewed frequently and it is desirable to use it more or less interchangeably with various types and brands of varnishes. Other things being equal, therefore, a varnish which makes use of a solvent characterized by known quality, ready availability and low price is the "best buy."

WHAT OF THE PROPORTION OF SOLVENT TO BASE? The solvent is merely the vehicle for transporting the load (base) to its proper destination. The vehicle is inherently much cheaper than the base. If the proportion of solvent to base is greater than that actually needed for transportation, one pays a high price for a cheap article. It is not a good buy.

The base is the part which remains in and on the windings after the solvent has evaporated. It is the part which serves the purposes for which insulating varnish is used. It constitutes the metamorphic "stone that bends." It is normally referred to as the "non-volatile" matter content of the varnish. It is the most expensive part, and if too little of it is contained in a gallon measure, more gallons are required for a given service.

For example, let us assume that we have bids at $1.50 per gallon from two different sources; the varnish offered in one case being *60 per cent non-volatile,* in the other case only *30 per cent non-volatile.* Assume 5000 gallons being purchased. Now we can figure our buy in either one of two ways, viz.:

(*a*) If we can use the heavier-bodied varnish *as is* (i.e., 60 per cent base) we need to purchase only 2500 gallons (instead of 5000). 2500 gallons at $1.50 equals $3750.00. Thus we have saved $3750.00 over what the 30 per cent non-volatile varnish would have cost us.

(*b*) On the other hand, it might be that the heavier-bodied varnish is too viscous (not the best consistency) to give us the penetration we need for our application. Then let us purchase, say, 1700 gallons of thinner at $0.10 per gallon; giving us 4200 gallons of 30 per cent varnish at a cost of $3920.00, or $0.93 per gallon instead of the $1.50 we would have paid had we accepted 30 per cent varnish at the price it was offered.

SHELLAC.

Courtesy Westinghouse Electric and Manufacturing Co.

Bug Harvests . . . Twice a Year.

Plant lice and shellac . . . they are part of the electrical industry. Shellac is used in some kinds of electrical insulating varnishes and it comes from bugs which feed on the sap of trees in India. These bugs or plant lice exude a resin that finally entombs them. Before the lice die their young are transplanted to other trees. The older generation and the resin it has produced is scraped from the trees. After much refining to separate resin and insect, the product is shellac of several grades. About six months are required for one crop of shellac.

This is purely a matter of purchase economy; other things being equal, therefore, the varnish containing the greatest proportion of base per gallon of measure, is the best buy.

WHAT PRICE LONGEVITY? This is a matter of purchase economy beside which the estimates given above on the relative costs of solvent and base are as nothing; the labor and overhead costs of putting varnish in an electrical winding, and solidifying it there, are many times the cost of the varnish used.

Roughly speaking, $1.00 worth of varnish will impregnate and protect the windings of $100.00 worth of electrical machinery, even when the machinery is newly built as a production line job. When it comes to repairs or renewals later, it may cost $1000.00 to disassemble, dry out, revarnish and reassemble a large generator or motor in service, using less than $25.00 worth of varnish.

For example, let us assume 15 years of reliable service to be striven for in a large generator or motor such as above mentioned. Let us assume further that 20 gallons of varnish are required for one repair job of this machine, and that we are offered one varnish of 3 years' life at a price of $1.00 per gallon, and another varnish of 15 years' life at the price of $5.00 per gallon. Just to make it simple (although it is not quite so simple or casual) assume that each one lives its allotted life, no more nor less than the 3 and 15-year periods, respectively; what does it add up to in the two cases?

3-Year Varnish.		*15-Year Varnish.*	
For varnish:		For varnish:	
$5 \times 20 \times \$1.00$	$= \$100.00$	$1 \times 20 \times \$5.00$	$= \$100.00$
For labor:		For labor:	
$5 \times \$1000.00$	$= 5000.00$	$1 \times \$1000.00$	$= 1000.00$
Total.....	$5100.00	Total.....	$1100.00

Thus the one with the shorter life would be the more expensive, in dollars alone, to say nothing of being five times as hazardous from the viewpoint of service reliability. This is in spite of the fact that the author assumed $5.00 per gallon for the better varnish

and $1.00 per gallon for the other. Actually, the difference in price per gallon is not that great, but it illustrates the point that the cost of varnish, per gallon, is of little importance.

This feature of purchase economy may, therefore, be summarized as follows: As soon as the varnish is put into place in the windings, it starts to die. When the point in its life has been reached where the varnish on the windings has cracked, it no longer serves the purpose for which it was placed there. A new application of varnish is then necessary. The longer the life of a varnish the fewer the applications that are necessary. Fewer applications mean fewer gallons purchased, less labor in applying it and fewer breaks in the periods of protection afforded to the windings. Other things being equal, therefore, the longest lived varnish is the best buy.

WHAT IS THE BEST BUY? The author is convinced that some form of work factor is the best means of evaluating this problem, but is not so sure about just how to do this; hence is withholding any definite suggestion for the present. At present he is merely attempting to take the subject apart. If he is able, eventually, with help, to put it back together again, he intends to offer just such an evaluation.

As to the Manner of Its Application.

Reverting back to the main question, it will be observed that the manner of application is concerned with six varnish characteristics, viz.: (1) flash point, (2) specific gravity, (3) viscosity, (4) draining, (5) chemical reaction, and (6) drying time. Let us examine each of these six factors as briefly as possible.

WHAT IS THE SIGNIFICANCE OF FLASH POINT? A low flash-point varnish has certain advantages over a high flash-point varnish; such as (a) a shorter drying time for the varnish, (b) less tendency to damage the insulation of enamelled magnet wire, and (c) probably better penetration of the varnish using the lighter solvents. Most manufacturers of electrical machinery use low flash-point varnishes for electrical windings.

To offset these advantages, a low flash-point varnish involves serious fire and explosion hazards. These hazards are recognized

by the machinery manufacturers and they are able to, and do, take the necessary safety precautions. In the naval service, however, and particularly aboard ships where most of the varnish is used, it is not practicable to take the same precautions that machinery manufacturers do in disposing of the solvent vapors. Consequently, the matter of low flash point is much more serious under shipboard conditions than on land.

When the Navy purchases and applies the varnish, it is concerned with the question of flash point. When machinery manufacturers apply the varnish, in their own works, to machinery destined for the Navy, it is of no interest to the Navy what the flash point is.

WHAT IS THE SIGNIFICANCE OF SPECIFIC GRAVITY? By definition, the specific gravity of a varnish is the ratio of its weight to that of water. It is not a measure by which the proportions of solvent or base, or by which the covering power of several varnishes can be compared. When the other characteristics of a given brand or type of varnish are definitely known, the specific gravity determinations indicate (within certain limits) if the varnish is of the proper "consistency" for best application to the windings. In short, it serves as a "control method," to insure uniformity of results as regards any one particular varnish, but does not furnish any safe comparison by which to judge different varnishes. Taken in conjunction with other control measures, it may be utilized as one step towards the assurance of product uniformity.

WHAT IS THE SIGNIFICANCE OF VISCOSITY? The chief value of a viscosity determination is likewise that of a control method; it is useful in comparing similar varnish samples, but it is not a measure of quality or suitability. The greater the viscosity, the greater will be the thickness of the varnish coating and the less will be its penetrative ability. In fact, viscosity in an insulating varnish intended for the impregnation of multi-layer windings, is something we have to take rather than something we seek or desire.

In this connection it might be pointed out that varnish at the time of its application, is a mixture or combination of somewhat diverse compositions in solution, rather than a single, definite chemical compound. Sudden changes in temperature or in the addition of

WORK FOR THOUSANDS.

Courtesy Westinghouse Electric and Manufacturing Co.

. . . Splitting Mica.

The mica fields of India make work for thousands of men, women and children. The mining of mica by natives is a haphazard operation, often dangerous. Irregular deposits of mica are followed in a series of tortuous, narrow burrows and holes. When water collects, it is bailed out by bucket brigades of women coolies. Mica is carried out in bundles and baskets on the heads of the workers . . . to huts where other coolies strip it into thin sheets by hand. From there it comes to us as material for insulation.

thinner, may throw parts of it out of solution. Also, the act of penetrating the minute capillary ducts or interstices of a multi-layer fine winding, tends to promote selective absorption; the less viscous portions tending to penetrate more deply than the heavier portions. In short, the windings act as a series of sieves, to trap the different fractions of the varnish at varying distances from the entering surfaces.

This is a phenomenon which is visible (as a change in color) to the eye, so it is known to occur. Just how to measure it in numerical terms, however, so that it may be evaluated, is something else again. Thus we lump all these variables in one grand indistinguishable mess, and try for the sake of product uniformity to use it as a control method.

WHAT IS THE SIGNIFICANCE OF DRAINING? As soon as windings which have been impregnated with varnish by immersion are taken out to permit the excess varnish to drain off, the changes which begin to take place are rapid and complex. The quantity of varnish which adheres to the windings at the start is governed by the original viscosity of the varnish and the speed with which the windings are withdrawn from the varnish bath. But as the draining proceeds, the solvent is evaporating rapidly and the varnish is becoming more and more viscous all the time. Eventually, enough solvent is evaporated to permit the varnish base to stop slipping and to stay put.

If the original viscosity is very low, and if the changes in the viscosity during draining are extremely rapid, it means that there will be a great difference in the film thickness between the top and the bottom portions of the windings. If the part happens to be an armature or other rotating part, too much of the varnish drains away from the top and too much collects at the bottom, thus throwing that part out of good mechanical balance. Sometimes mechanical means can be taken to turn such pieces over while they are draining and thus prevent such uneven deposits, but in other cases it is not practicable to do so. Such inequalities in the varnish filling and coating of windings also affects the thermal conductivities of the several paths with resultant inequalities of heat dissipation.

Unevenness in draining is a disadvantage from every viewpoint. In general, the more rapid the evaporation of the solvent the greater will be the difference in the film thickness; while heavier, less volatile solvents tend to give a longer flow and, therefore, greater evenness in film thickness. On the other hand, the heavier, less volatile solvents may lengthen the drying time and possibly damage enamelled magnet wire. It brings up the same old question again, of which feature shall be subordinated and how much? Varnish, like most other engineering matters, is just one long unhappy bit of dangling between the Devil and the Deep Blue Sea.

WHAT OF CHEMICAL REACTION? Since varnish plays the role of "guardian," it is essential that it does not react with or impair the physical and electrical properties of the materials of construction with which it is in contact. This applies to metals such as copper and iron, as well as to the various insulating materials. There is more or less so-called "acidity" in all vegetable oils used in insulating varnish; the degree varying with the nature of the oil used, the temperature and the degree of oxidation reached during drying, as well as the progressive oxidation which occurs in service. Reactions with metals produce salts which, even if they are non-conducting as long as there is no moisture present, will become conducting if the film cracks and permits moisture to enter. An electrolyte is formed.

Numerous studies have been made of the increase in acidity, with temperature and time (progressive oxidation), as displayed by a decrease in the insulation resistance, particularly with respect to insulating varnish and to varnished cambric using linseed oil varnishes. Many persons are no doubt familiar with the greenish blue soap which is sometimes visible next the copper conductors when old windings are dissected; such dissections representing post-mortems to acertain the causes of failures.

WHAT OF DRYING TIME? Drying time is important from the standpoint of production. It directly affects the rate of manufacturing production and also the length of time during which machinery undergoing repairs must remain idle. During the period of drying, the "sticky" windings take up dust and lint, making them more or less unsightly. Also if the coating on newly varnished rotating parts is not thoroughly dried before these parts

are put into operation, the varnish is thrown out into the air gap or on to other parts of the machine. Thus the rotating part has lost much of its protection while the ejected material gums up the air gap, reduces air circulation (if it gets into the air ducts) and has a very "messy" appearance in general. All in all, the effect is to so disgust the party making the repairs that he is very likely to overlook every other consideration and demand such a short drying time that only a "poor" varnish can meet it.

It is almost a truism with varnish that "quick drying" is "quick dying." If, therefore, everything else is to be subordinated to quick drying, there is not much to be gained by applying insulating varnish; it is dead almost as soon as it is applied. On the other hand, it is essential that the drying time be just as short as can be obtained and yet get varnishes with some life in them.

The atmospheric conditions aboard ship are not conducive to quick air drying at any time, and it does sometimes happen that machinery overhauls are made in certain localities during the "rainy" season. Thus a really good varnish might get a bad name for not drying, when a combination of circumstances was really at the root of the trouble.

Coupled with the foregoing is the fact that it is extremely hard to get consistent results on "drying time" even in a well equipped laboratory. There are so many variables that affect the result that it is almost impossible to control them all and to get the same result time after time. Some condition will alter slightly and affect the result beyond explanation.

As to the Services Rendered.

Again reverting to the main question and the outline thereunder, it will be observed that fourteen of the characteristics of any insulating varnish have to do with the services it is capable of rendering. Let us examine each of these fourteen points as briefly as possible.

WHAT OF PENETRATION? This is a highly important matter; for unless the varnish penetrates the veritable maze of wires and is carried to the very innermost recesses of the windings, it cannot really serve the purpose for which it is used. In spite of the attention which this feature merits, there is as yet no standardized test

for determining the characteristics of various varnishes in this respect. This situation will, no doubt, be corrected in due time, but how?

For years, the Navy has tried first one method, then another, to get the low down on this most elusive characteristic. Every method of test suggested by anyone else, as well as many ideas of its own, have been tried without success. Recently, a device which seems to give more promise than anything else to date has been worked out. The author wonders, however, just how long it will be before we find that it too has its shortcomings.

DEPTH OF DRYING. This also is a problem. The best insulating varnishes dry (harden) by oxidation and polymerization of the oils after the solvent has evaporated. This hardening is progressive throughout the life of the varnish. Oxidation begins at the *surface* and progresses inwards. The varnishes normally contain drying agents to accelerate this oxidation. If the drying action is too strong it may produce "surface drying," a hard outer film surface with soft, unoxidized varnish below. This is a very undesirable characteristic which should be guarded against. Undue emphasis placed on quick drying may introduce this evil.

Some bitter experiences have been had with certain very large, multi-layer coils, which cured hard on the outer surfaces and left the inner sections to stew in their own juices; until the acidity became so great as to cause heavy short circuits between the adjacent turns, in spite of the fact that these turns were separated by the usual textile covering. No amount of baking out was effective; because the crust merely became harder and the trapped solvent faced a progressively more impervious barrier against its escape.

Everyone can recognize at sight a deep drying varnish when it is cooked up in a shallow vessel; it looks very simple. It is an entirely different matter, however, to be able to simulate an actual winding with its labyrinthian maze of interstices, and to obtain *numerical* values. So far, we have not been able to do it, and we have only succeeded in making every suggested test look sorry when scrutinized in the cold light of factual data.

It is most important to know how to achieve such a useful yardstick, and we are certainly not giving up its search; but who has any new ideas to try out?

BINDING AND ADHESIVE PROPERTIES. The service rendered by varnish in this respect has already been touched upon briefly elsewhere herein. For some applications such as small, very high speed armatures, exceptionally good binding powers are required of the varnish used; this is to prevent such shifting in weight as would cause mechanical unbalancing and such weaving of "soft" (fine wire) coils as might result in damage thereto. It is also required that subsequent varnish applications adhere tenaciously to coats applied upon previous occasions.

As an example of what the binding or "bonding" properties of a varnish may be, consider the matter of laminated phenolic material. Here is a case where the bending stone principle, spoken of near the beginning of this article, has been sacrificed to obtain great bonding power. The very essence of laminated phenolic material is a series of *varnish* impregnated papers or fabrics bonded solidly together. In a coil winding we merely seek a slightly different balancing of the bending and the bonding properties which are normally in opposition.

Fortunately, this property is quite readily measured.

HIGH SOFTENING TEMPERATURES. It is essential that, after having once dried, the varnish does not soften sufficiently under the maximum operating temperature of the windings, to permit the coating to become sticky or collect dust; or, in the case of rotating parts, for the varnish to be thrown off or to change its position and so affect the mechanical balance of the rotor.

For certain large form-wound coils which must be held in stock for a considerable time before use, particularly coils which are for use as replacement parts or spares, it may be desirable to use a varnish which softens slightly upon sufficient heating, so that the coils can be formed to exact dimensions and readily assembled in the slots without tearing, cracking or other damage to the coil insulation. Ordinarily, however, any marked degree of thermoplasticity within the maximum operating temperature range for which the electrical windings are designed, is a fatal disadvantage to be guarded against.

CHEMICAL STABILITY OR PERMANENCE. The question of varnish "acidity" at the time of the varnish application has already been discussed in part under "as to the manner of its application." It

is just as important, however, that as a result of aging under the hot, humid conditions aboard ship, the heterogeneous compound known as varnish does not break down and form new combinations which may be deleterious to good insulation. In short, it is important that chemical stability under trying conditions be assured.

The term "permanence," to describe this goal of a fixity of characteristics over a long period of time, is just coming to the fore. It is a good word even if we do not as yet know just what it means, particularly as applied to varnish. The idea is the thing of importance, however, because if we cling to that we will soon be able to define it, which in turn will lead to the ability to measure; and when that much is done we will be at least halfway towards attainment.

DIELECTRIC STRENGTH. The service rendered by varnish in this respect has been briefly touched on elsewhere herein. Unfortunately it is very difficult to obtain consistent results in dielectric strength tests on any type of insulating material, even a homogeneous one; because there are too many variables which are extremely critical in determining just which way, and how, the juice is going to jump. It is not a question of the juice being temperamental, but rather that we are just not well enough acquainted with all its predilections.

The problem is further complicated by the fact that the service rendered in this respect is an indirect one, and, in order to simulate actual conditions, our methods of measurements must likewise be indirect. Varnish is used not alone but in combination with something else, and it not only acts differently in that combination than it does alone, but the other things in the combination also act differently by themselves than they do when combined with varnish.

When the elements in a combination are measured separately we can not add them up to arrive at their combined worth, nor can we subtract the value of one measured alone, from the combined value, to arrive at the worth of the other, alone. Even if we could, for one particular combination, it would not be the same for any other combination, because each influences the other to a different degree.

But the matter of trying to deduce the separate worths in a combination is not all there is to it. Neglecting all that part, consider a few lesser details; for example, it is very difficult to produce varnish films of anything like uniform thickness (see previous comments on "Draining"); the dust particles, air bubbles, etc., produce minute perforations in this film. Thus it can be appreciated that the chances of getting a poor film on the test specimen are very great. In many cases, therefore, the wide variations in the dielectric test data are really due to the variations in preparing the specimens rather than to variations in the actual quality of the varnish.

The degree of drying of the varnish film also vitally affects the dielectric strength test results. The degree of "acidity" in the same varnish changes with the degree of oxidation of the varnish film. During the process of oxidation, "low resistance" materials are formed in the varnish; these may increase up to a certain point during the drying process and then decrease as the drying is continued. Unless great care is exercised in drying all the test specimens in the same relative amount, the variations in the dielectric strength test results may be a measure of the particular stage of drying rather than of the dielectric quality of the varnish as fully dried.

The dielectric strength also varies with the film thickness; the value per unit of thickness decreases with increase in thickness. One reason is that the thinner the film the more advanced has the process of oxidation become at any given time within a comparatively short period after it appears to be dry. Another explanation is the commonly observed characteristic of most insulators to decrease in unit effectiveness with increased quantity; the dielectric strength-thickness curve is drooping, the particular characteristic depending upon the material under consideration.

Aside from the foregoing are the factors of temperature, relative humidity, electrode size, electrode pressure, electrode face surface, rate of application of voltage, voltage wave shape, which all affect the dielectric strength test results. In spite of all the precautions which are usually taken to eliminate such sources of error, variations in the final result still creep in. Dependence must, therefore, be placed in the average of a large number of tests rather than in the particular result of an individual test.

POWER FACTOR AND DIELECTRIC CONSTANT. POWER FACTOR. The power factor of insulating varnish in the dried film form is something of which the Navy has had no experience as yet, but which will be investigated. Some new power-factor testing equipment has recently been installed in the Material Laboratory, Navy Yard, New York, N. Y., for this purpose, among others. It is expected that power factor measurements may disclose interesting information as regards the development of new test methods to determine:

Resistance to water absorption.
Resistance to heat, and aging.
Permanence, stability, or whatever it is called.
Product uniformity.

DIELECTRIC CONSTANT. The dielectric constant is of importance (as a characteristic) for certain applications only. In general, however, and for the majority of its applications there is probably a close enough degree of similarity in this respect among the various brands of varnish, that there may be no need for prescribing any limits. Such measurements may, however, assist indirectly in explaining other observed phenomena, or assist in their interpretation.

INSULATION RESISTANCE. Theoretically, this is the characteristic which should give us the information we need about any insulating material. What would make more sense than to find out how high the resistance to the flow of electric current is in a material which is supposed to offer a high resistance? The very thought is basic to our conception and to our accepted definition of when a material is classed as an insulator and when as a conductor. Then what are we waiting for?

Frankly, the author does not have the spirit left to venture any more guesses, after having seen so many cherished ideas smashed without even a good explanation. It may be that it is like the door of a safety vault, with a dozen tumblers; and the door stays shut until every one is revolved to the exact degree and in the exact sequence. Figure out for yourself how many possible combinations you might have to try before the door falls open.

Everyone knows about the tumblers of *temperature, humidity, condition of equilibrium, filiamentary paths, electrode contact* and

all other such precautionary measures taken for the sake of consistency in the test results. Just what tumblers are missing? Do not misunderstand. The program of making *insulating resistance* tests is not being slackened any; one does not give up anything basic so easily; but so far the battle has gone against us. In time, it may turn, we hope.

WATER-PROOFNESS (FRESH WATER AND MOIST AIR). GENERAL. The moist atmospheric conditions obtaining aboard ship render this characteristic of very great importance. As has been previously pointed out, the exclusion of moisture is not only the primary reason for the use of varnish, but it is the hardest job that it has to fill. It ceases to fill that position in a satisfactory manner just as soon as the film cracks. Even before that stage is reached, however, the question of water absorption is raised.

METHODS. The resistance which varnish offers to the entrance of moisture is usually determined in two ways. One test is known as the "water absorption" test and the other as the "water-proofness" test. Neither test seems to quite fill the bill, but in the absence of something better it is necessary to make use of both of them. Still other methods are being or will be tried in an effort to get to the bottom of this matter.

WATER ABSORPTION. The main objection to this method is that it is subject to all the vagaries of the dielectric strength test, plus a few of its own. If the film thickness and the degree of oxidation vary as between the different specimens, the test results may be so inconsistent as to be almost valueless. Take, for example, two specimens from the same varnish and let one film be 4 mils thick and the other 6 mils thick. Assume that both films are equally uniform in texture and hardness so that the depth of penetration of water is the same in both cases. If this depth be assumed to be 2 mils, then the unpenetrated film thickness will be 2 mils on the first specimen and 4 mils on the second. The effective thickness of the first film is only about 50 per cent that of the second film, but its measured thickness is 66⅔ per cent. Dividing the total voltage required to puncture the *effective* thickness by the *measured* thickness, gives a big variation in the unit dielectric strength of the two specimens. Yet the water absorption was exactly the same in both cases; the example was predicated on that assumption.

One theory is that the reduction in dielectric strength due to the immersion of the specimens in water is a measure of the absorption. And so it might be if the film thickness and the several other factors could be held constant. There is no question about the value of the test, it is merely a question of being able to control the test condition and to interpret the test results so as to be able to render consistent decisions.

WATER-PROOFNESS. By this method of determining the resistance offered by varnish to the entrance of water, an attempt is made to measure the length of time it takes for the water to completely penetrate the varnish film. The resistivity of the film is gradually lessened, through long immersion in water until it reaches such a low ohmic value as actually to become a fair conductor at low voltages. The test results are quite consistent, but the period required for testing is very long and there is some question as to what the results really signify in terms of operation under the usual service conditions.

Of the two above methods of approach the first is the more directly applicable to service conditions, but the results are often so erratic as to impair their usefulness. The second method is the more consistent, but it is very much longer and in the end probably does not mean nearly so much as the first one.

SEA WATER-PROOFNESS. Everything discussed above with respect to fresh water and atmospheric moisture, plus a lot more, applies to the question of sea water. Sea water gets into insulation as a result of " flooding," or " spray," or as fine globules suspended in the air and wafted about, even in those locations aboard ship which are normally protected against what might be termed actual " spray." And sea water has just about everything that one desires protection against in his search for permanence in electrical insulation.

All the metallic elements in the earth's crust are represented in sea water, but not in the innocuous forms of oxides and silicates comprising the rock on land. Once sea water gets into the interstices of windings and of solid insulation (by absorption) these electrolyte producing compounds stay there. The insulation may be dried out insofar as the H_2O is concerned, but the salt is de-

posited in the internals, where it merely has to lie in wait for moist air in order to become sea water (and electrolytic) all over again.

The measurement of this characteristic follows the same general procedures outlined for water-proofness, for alkali-proofness and for acid-proofness. Fortunately, it is easy to simulate service conditions in a laboratory test, and it leaves no doubts, nor any need for circuitous deductions or interpretations as to what happens. The principal difficulty is to obtain enough values to be able to express them numerically.

ACID-PROOFNESS. The resistance of varnish to acids is of importance only in particular installations. Sulphuric acid is, in general, the only one with which naval shipboard installations are concerned. No particular difficulty has been experienced in obtaining insulating varnishes of sufficient acid resistance to meet the usual service conditions.

ALKALI-PROOFNESS. This is a characteristic which may be of importance for certain industrial applications on land, but which does not, in general, concern the Navy. Resistance to sea water is, of course, essential, but tests to determine this characteristic are covered under " sea water-proofness " above.

OIL-PROOFNESS. The resistance of insulating varnish to lubricating oils and greases is of very great importance in all electrical machinery installations. The spilling of oil on the electrical windings due to carelessness in filling oil reservoirs, to undetected overfilling, to defects in the oil piping, and to the shaft " slinging oil," is a more or less frequent occurrence in practically all machinery installations. In spite of elaborate precautionary instructions intended to prevent such a condition, this is the situation which exists in practice and has to be recognized. One of the main purposes of varnish is to prevent such oil from doing any damage.

In the case of shipboard installations, the rolling and pitching of the vessel greatly aggravates this situation. With motor and generator shafts operating at various angles of inclination, the tendency to " sling oil " over the armature, and its later collection in the lower portion of the magnet frame, results in the oil-soaking of both the rotating and the stationary electrical windings. This situation is particularly prevalent on submarines.

In general, the varnishes containing the higher percentages of drying oil are the most oil proof. The oil resistance of any varnish is increased, however, by overbaking; in fact, the varnish can be made to appear as an oil-proof one when it really is not, simply by baking it until the film is so hard that it is practically unaffected by the original solvent. To do this, however, is sacrificing its life, its water-proofness (due to cracking), its acid-proofness, etc. This brings out the point that, to be fair, tests for oil proofness should be made on samples which have been baked to a uniform degree of hardness in relation to their surface drying times. It also illustrates anew the opposing characteristics of varnish.

LIFE. Reference to a previous section of these notes will indicate the stress which has already been laid upon this characteristic. The main function of varnish is served only so long as the varnish is *alive*. The span of life of a varnish is short at best. The higher the temperature to which it is subjected, the shorter its life. Since the question of longevity is of such vital importance, and since temperature is such a large factor in that longevity, it is but natural that attempts should be made to use the one as a measure of the other.

While the longevity of a varnish under the high temperature endurance test is not a true measure of its longevity under actual service conditions (because of the different temperatures involved), it is a proper measure of the *relative longevity* of two or more brands of varnish under actual service conditions. In other words, while a high temperature endurance of 100 hours does not mean a service life of 100 hours, a high temperature endurance of 100 hours for one brand means a probable service life of twice that of another brand having a heat endurance of only 50 hours.

The high temperature endurance test is a measure of the length of time (in hours) that a varnish film may be baked at a given temperature (100 degrees C.) before it becomes so brittle that the film will crack when it is bent around a mandrel of a given diameter. In this test it is essential that all specimens be cooled to the same temperature and bent around the mandrel in a uniform manner. If the conditions (temperature, tension, film thickness, mandrel diameter, speed of bending, etc.) are not maintained uniformly, the results will not be fairly comparable.

WHAT NEXT?

The author recognizes the fact that he started out to ask questions, and that he appears to have wound up by answering many of them instead. This was certainly not the real intent; and if it appears that way he asks the reader to consider any statements made, no matter how emphatic they may appear, as *questions* not *answers*. For the author himself does question them, and he invites that questioning attitude on the part of others.

The reason, however, for not changing the arrangement of this discussion so as to make it all either one thing (questions) or the other (answers) is a sentimental one. Part of this article was written in 1923, another part in 1932 and these two parts have been woven together in this year of 1940. If it means anything to the reader, the *questioning* spirit came last, having been in the process of developing for that period of time.

In the interim between the start and the conclusion of this article, several hundred complete tests on insulating varnish have been made at the Material Laboratory, Navy Yard, New York, N. Y. In view of the cooperative spirit displayed by the various manufacturers of insulating varnish, in submitting their wares for such tests, and in view of the extent to which their representatives have confided their own aims, hopes and aspirations on this subject, it is hoped that a complete tabulation of these test results (manufacturers' and brand names deleted) can be published in a subsequent article.

In the meantime, two little straws may serve to show which way the wind is blowing, and what may be expected in the period that lies ahead:

(*a*) 1923. We had to be satisfied with a life of 50 hours at 105 degrees C., for a black insulating varnish that could be dried in a short enough time for naval ships and stations to use satisfactorily.

(*b*) 1940. While such a value is not specified as yet, there are now available, black insulating varnishes with a life of over 3500 hours at 105 degrees C.

Not even a preliminary answer to the question of "what next" is offered at this time; but, if it should be that the art of manufac-

turing insulating varnish keeps up the pace in the future that it has set in the past, one might turn an old familiar phrase around to read " while there is *hope* there is *life* "; this, in turn, can be shifted back to the original objective by stating that " when we have *enough* life to *spare,* over and beyond that required by the *first horsepower,* we can put it into *another* horsepower; and then we may have *two* horsepower instead of one.

PRESSURE DROP TESTS ON GLOBE AND ANGLE VALVES WITH OIL AND WATER FLOW.

By Milton Karr * and Loring W. Schutz.†

The data for this paper was originally obtained for two theses: One written in 1937 by Milton Karr and Harry D. Crapon, and the other in 1938 by Loring W. Schutz and Archibald M. Main, Jr., for the degree of Bachelor of Science in Naval Architecture and Marine Engineering at the Massachusetts Institute of Technology.

The paper is published for several reasons: First, it illustrates the thorough and workmanlike method of doing a job that results from contemporary technical schooling. Again, it emphasizes the complex problem which is involved in warship construction, even in the disposition of a small brass valve. The data itself increases that already available for solving the complex problem.

The design of ship piping lends itself to difficulties because of the large number of fittings in a relatively short total length of pipe. In the usual piping problem, such as that met by the civil or mechanical engineer, the pressure drop through fittings may often be but a fraction of the total head loss of the system. However, the marine engineer, designing under limitations of space and weight, often finds the pressure drops through fittings to be a large part of the total loss in the system.

In the past, data available to the marine engineer has not proved entirely satisfactory for the solution of the particular problem. With this in mind, tests were conducted on two common fittings used in ship's piping in an effort to determine the head loss produced under specific conditions of flow, density, diameter, and viscosity as expressed by Reynolds Number.

* Instructor of Marine Engineering at the Massachusetts Institute of Technology.
† Junior Naval Architect, Navy Yard, New York.

The valves tested were as follows:

1. Globe valve—½-inch, 200-pound brass, screwed Navy type.
2. Globe valve—1½-inch, 200-pound brass, screwed Navy type.
3. Angle valve—1½-inch, 200-pound brass, screwed Navy type.

Results.

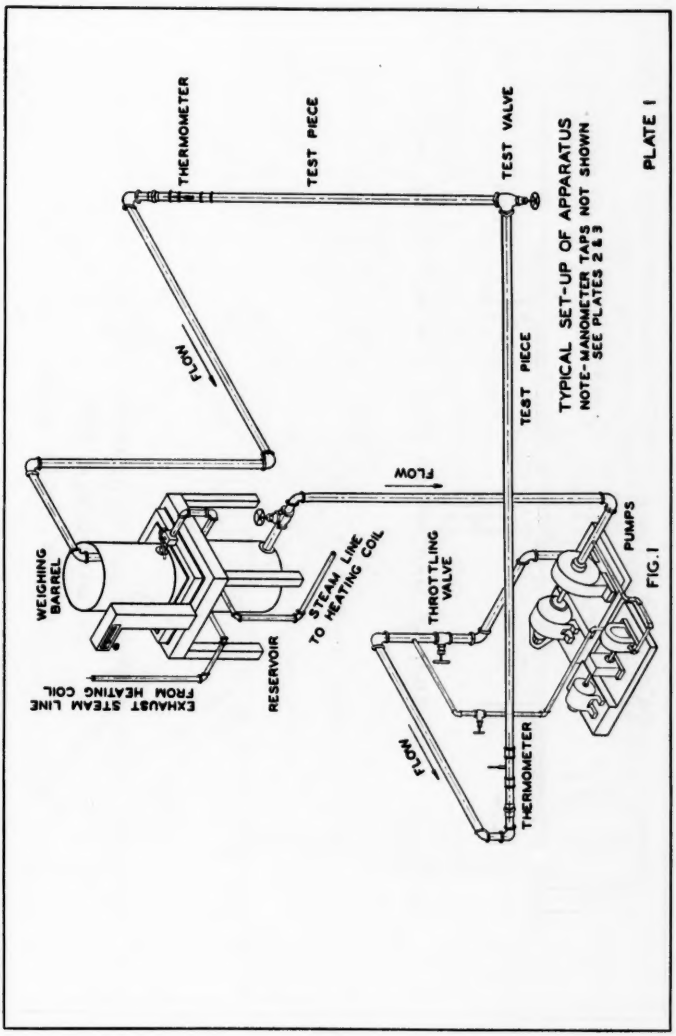
The data from these tests were plotted in terms of Reynolds Number and head loss in equivalent length of pipe in pipe diameters.

The three curves show three common characteristics:

1. The head loss increases rapidly from the lower Reynolds Numbers to a critical point.
2. A critical is reached at a Reynolds Number of approximately 2000.
3. Beyond the critical, the head loss again decreases, rapidly at first, but with a tendency toward a decreasing slope as the Reynolds Number increases beyond 50,000.

Description of Apparatus.

The apparatus consisted of a cyclical system in which the test valve could be mounted and the pressure drops across various points observed. Plate 1 shows a typical arrangement for conditions of oil flow. The oil, Socony-Vacuum No. 1800, was stored, and heated by a steam coil, in a seventy-five-gallon barrel. The pumps moved the oil through the valve, which was mounted between two lengths of pipe, on which were located the taps for taking off pressure readings. For the ½-inch globe valve, two six-foot lengths of ½-inch brass pipe were used; two eighteen-foot lengths of 1½-inch seamless drawn steel tubing were substituted for the 1½-inch valve tests. Plates 2 and 3 show the location and the details of fitting the manometer taps. Each tap consisted of a ½-inch standard pipe tee or 90-degree elbow brazed over a .188-inch diameter hole drilled on the horizontal diameter of the pipe. The inner wall of the pipes was carefully inspected after brazing for burrs or eruptions. At the up and down stream ends of the test pieces, thermometer wells were fitted as shown in Plates 2 and 3.


THERMOMETER
TEST PIECE
TEST VALVE
TYPICAL SET-UP OF APPARATUS
NOTE-MANOMETER TAPS NOT SHOWN
SEE PLATES 2 & 3
PLATE I
FLOW
TEST PIECE
FLOW
WEIGHING BARREL
STEAM LINE TO HEATING COIL
THROTTLING VALVE
PUMPS
FIG. I
EXHAUST STEAM LINE FROM HEATING COIL
RESERVOIR
FLOW
THERMOMETER

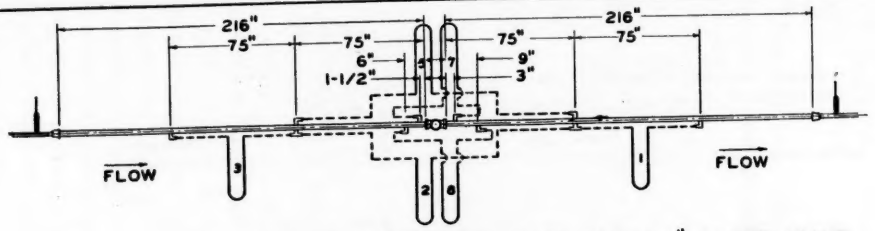


FIG. 2 LOCATION OF MANOMETER TAPS FOR 1-1/2" GLOBE VALVE

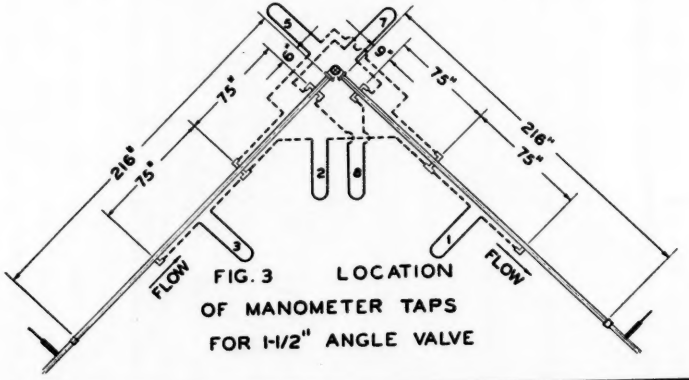


FIG. 3 LOCATION OF MANOMETER TAPS FOR 1-1/2" ANGLE VALVE

PLATE 2

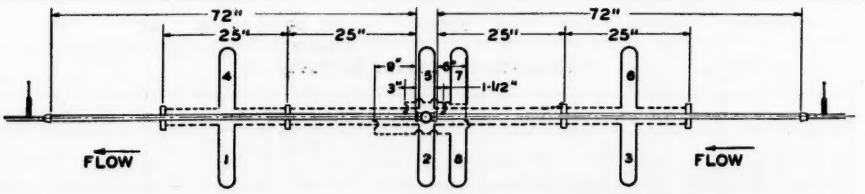


FIG. 4 LOCATION OF MANOMETER TAPS FOR 1/2" GLOBE VALVE

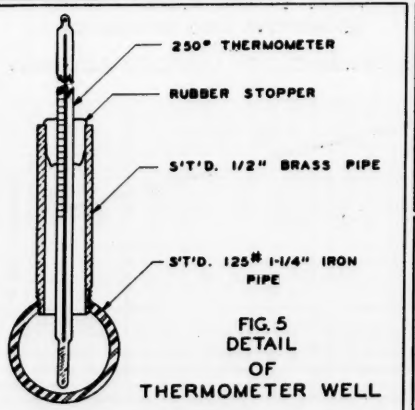


FIG. 5
DETAIL
OF
THERMOMETER WELL

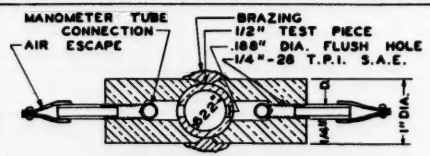


FIG. 6 DETAIL OF 1/2" PIPE
MANOMETER ATTACHMENT

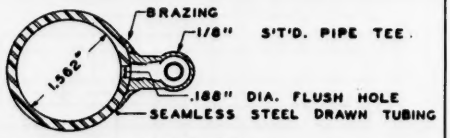
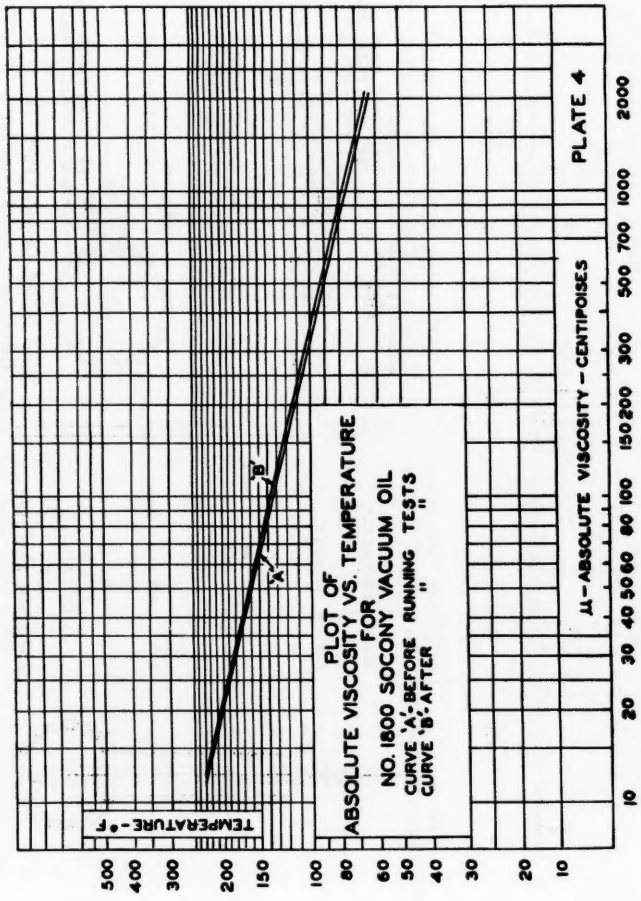


FIG. 7 DETAIL OF H/2" PIPE
MANOMETER ATTACHMENT

PLATE 3


PLOT OF
ABSOLUTE VISCOSITY VS. TEMPERATURE
FOR
NO. 1800 SOCONY VACUUM OIL
CURVE 'A'- BEFORE RUNNING TESTS
CURVE 'B'- AFTER "
"
TEMPERATURE - °F
500
400
300
200
150
100
80
60
50
40
30
20
10
A
B
μ - ABSOLUTE VISCOSITY - CENTIPOISES
10 20 30 40 50 60 80 100 150 200 300 500 700 1000 2000
PLATE 4

Pressure drops were obtained by mercury manometers, provided with air vents, and centrally located between tap points. Reference to Plates 2 and 3 show that drops across the following regions could be observed.

1. Valve only.
2. Valve, plus up and down stream length of pipe.
3. An up and down stream length of pipe only.

The oil, after passing through the test pieces and valve, was discharged into a weighing barrel, where the flow rate could be determined over any desired period of time. It was then returned to the storage barrel.

ABSOLUTE VISCOSITY OF WATER
REF. INTERNATIONAL CRITICAL TABLES

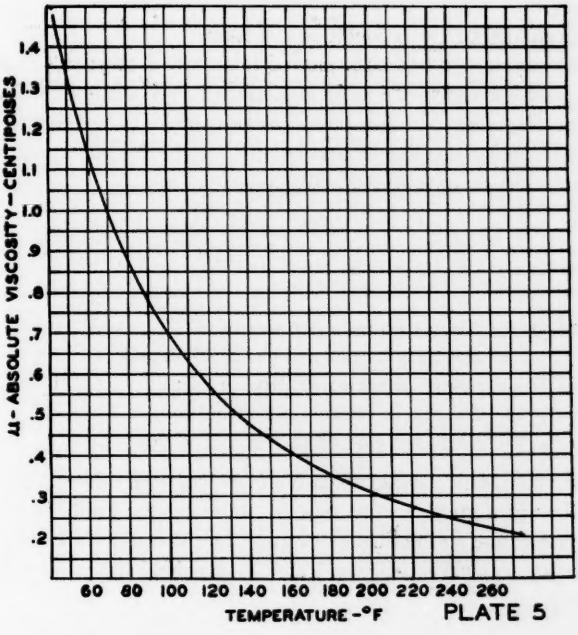


PLATE 5

For conditions of water flow the apparatus was essentially the same. However, heating was eliminated in the tests on the angle valve because of the availability of larger capacity pumps. The viscosity of the tap water used was taken from the International Critical Tables. (See Plate 5.) The oil, which was heated in an open tank, was subject to viscosity changes. To obtain an average figure for the viscosity, tests were conducted before and after the investigation, using a Saybolt Universal Viscosimeter under A. S. T. M. specifications. The result of these tests are shown on Plate 4.

Description of Tests.

The globe valve was tested with the pressure over the disc. No tests were run with pressure under the disc.

The angle valve was tested with the pressure both over and under the disc. Both valves could be packed either open or closed.

The operation of the apparatus consisted of pumping the fluid through the system, and observing the several pressure drops which corresponded to given conditions of flow.

The data taken has been plotted in terms of Reynolds Number and head loss expressed as an equivalent length of pipe.

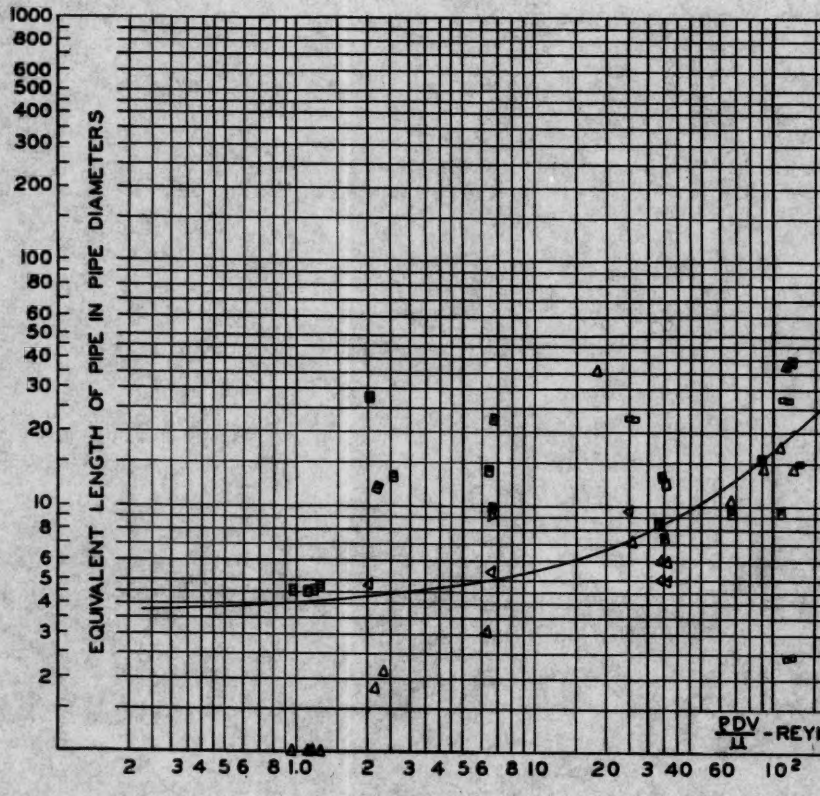
The Reynolds Number offers the advantage of a dimensionless expression for the flow characteristics of a fluid moving through a pipe when the proper units are taken.

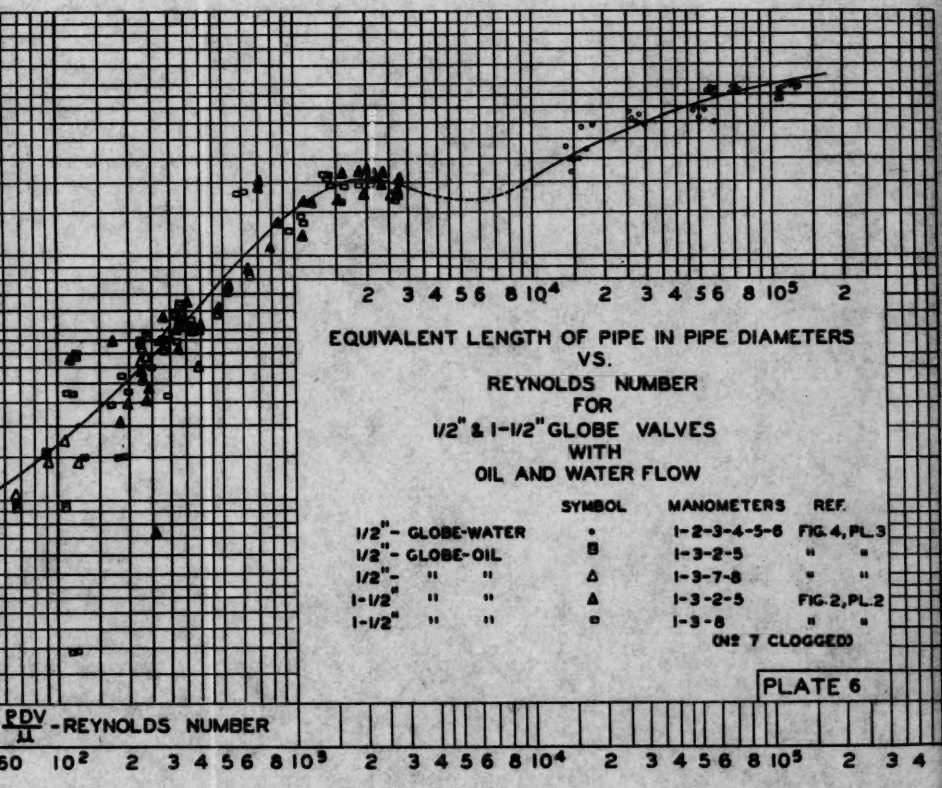
$$R_e = \frac{\rho DV}{\mu} = \frac{\rho D}{\mu} \times \frac{Q}{\frac{\pi D^2}{4}\rho} = \frac{4Q}{\mu \pi D}$$

where Q = flow in pounds per second
μ = viscosity in pounds per foot-second
D = inside diameter of pipe in feet
ρ = density in pounds per cubic foot
V = average velocity in feet per second

For these tests the Reynolds Number was easily determined for any flow in a given size of pipe by weighing the fluid discharged in a given time and noting its average temperature. The temperature-viscosity curves (Plates 4 and 5) provided the proper figure for viscosity.

XU


EQUIVALENT LENGTH OF PIPE IN PIPE DIAMETERS
1000
800
600
500
400
300
200
100
80
60
50
40
30
20
10
8
6
5
4
3
2
2 3 4 5 6 8 1.0 2 3 4 5 6 8 10 20 3 40 60 10²
PDV/μ - REY


2 3 4 5 6 8 10⁴ 2 3 4 5 6 8 10⁵ 2
EQUIVALENT LENGTH OF PIPE IN PIPE DIAMETERS
VS.
REYNOLDS NUMBER
FOR
1/2" & 1-1/2" GLOBE VALVES
WITH
OIL AND WATER FLOW
SYMBOL MANOMETERS REF.
1/2"- GLOBE-WATER • 1-2-3-4-5-6 FIG. 4, PL. 3
1/2"- GLOBE-OIL B 1-3-2-5 " "
1/2"- " " Δ 1-3-7-8 " "
1-1/2" " " ▲ 1-3-2-5 FIG. 2, PL. 2
1-1/2" " " ▫ 1-3-8 " "
(№ 7 CLOGGED)
PLATE 6
DV/μ - REYNOLDS NUMBER
50 10² 2 3 4 5 6 8 10³ 2 3 4 5 6 8 10⁴ 2 3 4 5 6 8 10⁵ 2 3 4

From the expression $R_e = \frac{4Q}{\mu\pi D}$ it may be seen that the Reynolds Number could be varied in four ways, namely:

1. Variation of pump discharge.
2. Variation of fluid.
3. Variation of fluid temperature.
4. Variation of pipe size.

For Reynolds Numbers up to approximately 3000, the head losses were obtained with oil flow; beyond this point, water was substituted for the oil.

The head loss, read on the manometers in inches of mercury is expressed as an equivalent length of pipe in pipe diameters (see Appendix I).

At the lowest Reynolds Numbers, the manometer readings were exceedingly small, and the errors in reading them led to a scattering of points. Between the Reynolds Number of 3000 to 8000 no points were obtained due to the limitations of the apparatus.

Beyond the Reynolds Number of 8000, the oil was replaced by water.

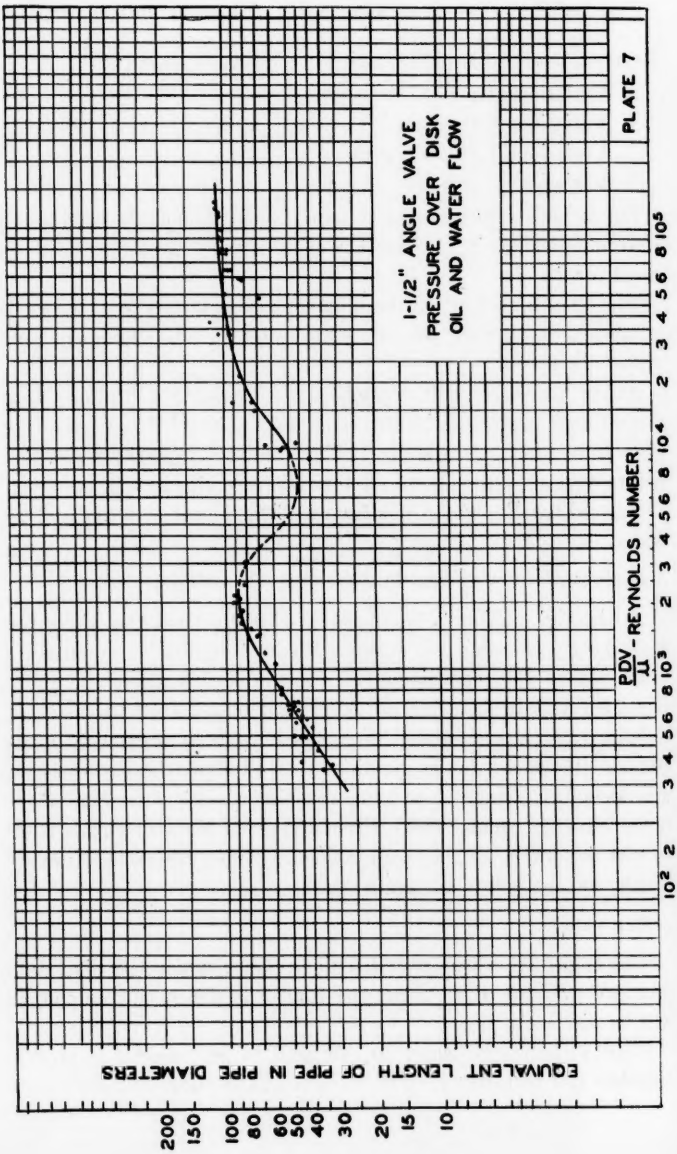
GLOBE VALVE: (See Plate 6) (Pressure Over Disc).

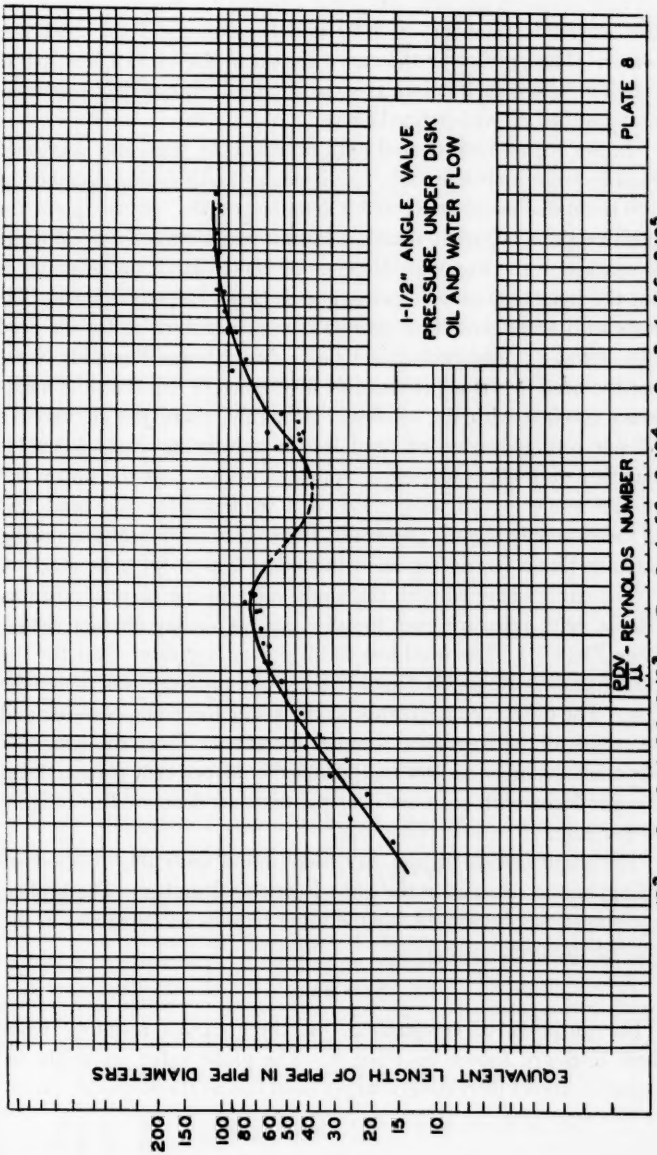
At the high Reynolds Numbers of 15,000 to 100,000, where the fluid used was water, the curve is distinctly defined as increasing from 250 to 450 pipe diameters. In the region where the curve is shown as a broken line no data could be obtained to justify a definite trend. The reasons for this have been previously mentioned. However, under conditions of oil flow, the curve shows a tendency to change slope after reaching a critical at about Reynolds Number equals 2000. From Reynolds Number equals 2000 to 200 the curve is again well defined. Below Reynolds Number equals 200 the points are again scattered.

ANGLE VALVE:

The angle valve was tested with pressure under the disc, and pressure over the disc. Since the former is the more common, it will be discussed first.

Plate 8 shows the range investigated to be between Reynolds Number 100 to 100,000. The portion of the curve obtained with


1-1/2" ANGLE VALVE
PRESSURE OVER DISK
OIL AND WATER FLOW
PLATE 7
EQUIVALENT LENGTH OF PIPE IN PIPE DIAMETERS
200
150
100
80
60
50
40
30
20
15
10
PDV/μ - REYNOLDS NUMBER
10² 2 3 4 5 6 8 10³ 2 3 4 5 6 8 10⁴ 2 3 4 5 6 8 10⁵


1-1/2" ANGLE VALVE
PRESSURE UNDER DISK
OIL AND WATER FLOW
PLATE 8
EQUIVALENT LENGTH OF PIPE IN PIPE DIAMETERS
200
150
100
80
60
50
40
30
20
15
10
DV/ν - REYNOLDS NUMBER
10² 2 3 4 5 6 8 10³ 2 3 4 5 6 8 10⁴ 2 3 4 5 6 8 10⁵

oil flow is shown between Reynolds Number of 100 to 2500. From the Reynolds Number of 8000 and beyond, the fluid used was water. The portion of the curve shown in the broken line marks a region where such points as were obtained were widely scattered, and justified no well-defined curve.

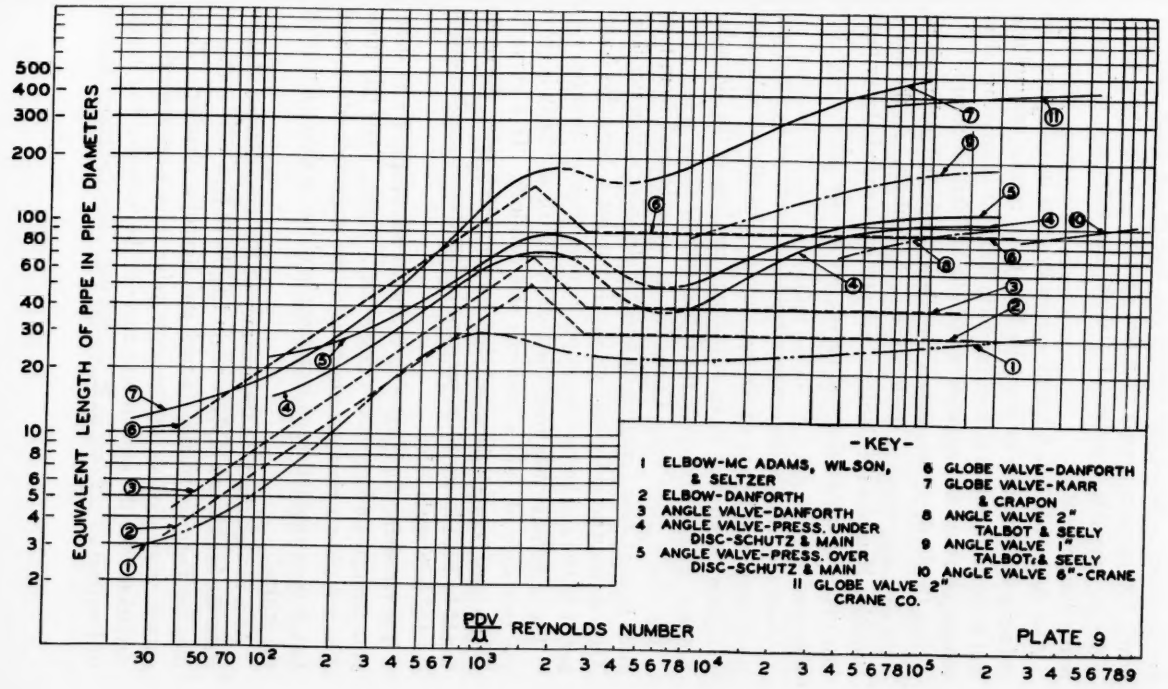
In the region investigated with oil flow, the head loss increased rapidly from approximately 15 diameters at Reynolds Number of 100 to 50 diameters at Reynolds Number of 70. At this point the positive slope begins to decrease until a maximum of 72 diameters is reached at the Reynolds Number of 1800. Here the tendency is for the slope to become negative and the head loss to diminish. The tests with water flow show again a rising characteristic for the head loss, but at a considerably lesser rate. At the lower Reynolds Number the head loss is approximately 30 diameters less than the maximum reached with the oil flow. Roughly, there has occurred a 50 per cent reduction of head loss in advancing from Reynolds Number of 1800 to Reynolds Number of 8000. At the upper end of the range investigated, there is a tendency for the head loss to increase only slightly; however, there were no indications during the tests that a second region of instalibity was about to be reached.

The curve obtained with the angle valve in the reverse position, that is, with pressure over the disc, shows similar characteristics. (See Plate 7.) The head loss at all points is greater than for the corresponding Reynolds Number when the pressure was under the disc. The difference is greatest at the lower Reynolds Number and decreases as the Reynolds Number increases. Over the range of Reynolds Number of 100 to 100,000 the 33 per cent greater head loss produced with pressure over the disc has decreased to roughly 20 per cent.

The same critical region has been found with this method of fitting, and at practically the same Reynolds Number. The broken line shown has been used for the same reasons as explained in the case of the previous curve.

Summary.

For purposes of comparison, the three curves resulting from these tests are shown on Plate 9. The globe valve, as might be expected, shows increasingly larger head loss as the Reynolds Num-


EQUIVALENT LENGTH OF PIPE IN PIPE DIAMETERS
500
400
300
200
100
80
60
40
30
20
10
8
6
5
4
3
2
30 50 70 10² 2 3 4 5 6 7 10³ 2 3 4 5 6 7 8 10⁴ 2 3 4 5 6 7 8 10⁵ 2 3 4 5 6 7 8 9
PDV/μ REYNOLDS NUMBER
-KEY-
1 ELBOW-MC ADAMS, WILSON, & SELTZER
2 ELBOW-DANFORTH
3 ANGLE VALVE-DANFORTH
4 ANGLE VALVE-PRESS. UNDER DISC-SCHUTZ & MAIN
5 ANGLE VALVE-PRESS. OVER DISC-SCHUTZ & MAIN
6 GLOBE VALVE-DANFORTH
7 GLOBE VALVE-KARR & CRAPON
8 ANGLE VALVE 2" TALBOT & SEELY
9 ANGLE VALVE 1" TALBOT & SEELY
10 ANGLE VALVE 6"-CRANE
11 GLOBE VALVE 2" CRANE CO.
PLATE 9

ber increases. Below the critical the loss in the globe valve is nearly 100 per cent greater than that developed in the angle valve, while at Reynolds Number of 100,000 the difference in loss is nearly 400 per cent.

Also shown on Plate 9 are the curves by other investigators. In reducing these results to the units here used, certain assumptions had to be made due to lack of complete data. It is necessary then to recognize that discrepancies can arise out of this procedure.

The Danforth curves, shown for elbow, angle and globe valves, show close agreement in the location of the criticals. At the lower Reynolds Numbers the Danforth line predicts a slightly lower head loss than the angle valve test curve, but for the globe valve the difference varies.

Above the criticals, the constant head loss predicted by Danforth is not supported by any of the test curves. The elbow angle valve and globe all agree as to an increasing head loss as the Reynolds Number is increased beyond the critical region.

The Crane globe valve Curve No. 11 was obtained from tests made by the Crane Company on 2-inch globe valve with water flow. This curve supports the authors' conviction that at Reynolds Number of 50,000 and beyond the slope of curve tends to decrease.

The angle valve data from the tests by Talbot and Seely is given for two sizes of valves, one-inch and two-inch. If a curve is interpolated between these two sizes for a 1½-inch valve, the result would be head losses 23-33 per cent higher than those obtained in the 1½-inch angle valve.

The remaining angle valve curve shown was obtained with steam flow through a 6-inch valve (Crane Co.). Again, while this curve does not show a close agreement as to the magnitude of the head loss, it does substantiate an increasing head loss in the region above the critical.

As a further check on the accuracy of the data obtained in these tests, the head losses in the straight lengths of pipe were converted into the Fanning friction factors and Reynolds Number. These curves, when plotted for the angle valve tests, gave a maximum deviation from the smooth tube curve in the turbulent flow region of less than 10 per cent at $R_e = 200{,}000$. For the lowest R_e, the

data taken with pressure over the disc gave a difference of 17 per cent from the Poiseuille line. For the pressure under the disc, the difference was 8 per cent or less. In the case of the globe valve tests, the agreement was even better.

Appendix I.

Calculation of Pressure Drop Across Valve in Pipe Diameters.

The authors used the following method for obtaining the pressure drops across the valve in terms of equivalent length of pipe in pipe diameters:

½-inch globe valve. (See Plates 3 and 6.)

Manometer taps are spaced twenty-five inches or fifty nominal diameters.

h_1, h_3, h_4 and h_6 equal drop in inches of mercury across fifty nominal diameters of straight pipe.

h_2 and h_5 equal drop in inches of mercury across valve plus one hundred nominal diameters of straight pipe.

Drop across valve in inches of mercury equals h_v equals

$$\frac{h_2 + h_5}{2} - \left(\frac{h_1 + h_4}{2} + \frac{h_3 + h_6}{2}\right)$$

Average drop in inches of mercury across fifty nominal diameters of straight pipe equals h_p equals

$$\frac{h_1 + h_4 + h_6 + h_3}{4}$$

Inside diameter of one-half-inch brass pipe equals 0.622 inches.

Fifty nominal diameters are equivalent to $\frac{25}{.622} = 40.19$ inside diameters.

Drop across valve in pipe diameters equals $\frac{h_v}{h_p} \times 40.19$.

Appendix II.

SAMPLE CALCULATION.

Data:

Water flow	4000 pounds per hour.
Water temperature	70 degrees F.
Pipe	10-foot, ½-inch brass.
Fittings	2 globe valves. 1 angle valve.

CALCULATION OF REYNOLDS NUMBER:

Assume specific gravity of water equals 1.0. Angle valve fitted with pressure over disc.

For Reynolds Number R_e to be dimensionless, μ must be expressed in pounds per foot-second if the diameter is expressed in feet and the flow in pounds per second.

$$R_e = \frac{\rho DV}{\mu} = \frac{4\rho DQ}{\mu\rho\pi D^2} = \frac{4Q}{\mu\pi D}$$

$\mu = .98$ centipoises. (See Plate 5.)

$$R_e = \frac{4 \times 4{,}000}{3{,}600} \times \frac{1}{\pi} \times \frac{1}{.98 \times .000672} = 41{,}300$$

CALCULATION OF PRESSURE DROPS:

Referring to Plate 6—at $R_e = 41{,}300$ pressure drop for globe valve in equivalent length of pipe = 380 pipe diameters:

$$\text{Equivalent length of pipe} = 380 \times \frac{.622}{12} \text{ or } 19.7 \text{ feet of } \tfrac{1}{2}\text{-inch}$$

nominal diameter brass pipe.

For two valves, equivalent length = 39.4 feet.

Referring to Plate 7—at $R_e = 41{,}300$ pressure drop for angle valve in equivalent length of pipe = 98 pipe diameters:

$$\text{Equivalent length of pipe} = 98 \times \frac{.622}{12} \text{ or } 5.07 \text{ feet of } \tfrac{1}{2}\text{-inch}$$

nominal diameter brass pipe.

Total equivalent length of fittings = 39.4 + 5.07 = 44.47 feet.

Equivalent length of system using " no-length " concept = 10 + 44.47 = 54.5 feet.

Pressure drop through pipe line and fittings using Fanning's equation

$$p = \frac{.323\ f\ l\ S\ V^2}{D}$$

where

p = pressure drop, pounds per square inch.
f = Fanning's friction factor.
l = length of pipe, feet.
s = specific gravity of fluid.
V = average velocity of fluid, feet per second.
D = inside diameter of pipe, inches.

$$V = \frac{Q}{A\rho} = \frac{4{,}000}{3{,}600} \times \frac{4}{\pi \times \left(\frac{.622}{12}\right)^2} \times \frac{1}{62.4} =$$

8.45 ft./sec.

$$p = \frac{.323 \times .0055 \times 54.5 \times \overline{8.45}^2 \times 1}{.622} = 11.1 \text{ lbs./inch}^2$$

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WATER CIRCULATION IN A-TYPE MARINE BOILERS.

By E. P. Worthen.*

Progressive increase in the permissible rating of equipment units follows after the discovery and cure of limiting factors in design. These factors usually appear one at a time in succession as the previous limit is lifted. It is well known that circulation in A-type boilers has proved adequate for rates of combustion up to about one pound of oil per square foot of boiler water heating surface and to about forty-five pounds of oil per square foot of radiant surface. Above these rates, circulation tends to become inadequate without special precautions. In the past, factors such as furnace design, burner design, and the type of fuel have all tended to make the limit of forcing come within the rate for which circulation was adequate. At present, these other limitations have been overcome. Efficiency can be obtained with a small boiler at high rates by using economizers and air heaters. Furnaces and burners have ample capacity. But a greater steam output may be obtained only by increased water circulation. The boiler rating is determined by the heat input which the tubes can absorb.

For the last forty years the Yarrow or A-type boiler has been preeminently successful for naval service where there has been a great demand for small size and low weight for a given steam quantity. In the face of such continuous good performance there can be no doubt that this type is fundamentally sound. This paper has been written in an attempt to determine if possible those inherent fundamental characteristics which make the A-type boiler essentially capable of producing good water circulation.

* Fore River Plant, Bethlehem Steel Corporation, Shipbuilding Division. In charge of heat transfer equipment design.

At this point a brief discussion of the important factors governing circulation is necessary. Only the following will be discussed in this paper:

1. Separation in steam drum.
2. Separation in water drum.
3. External downcomers.
4. Depth of tube bank.

Certain factors have not been mentioned and a word of explanation should be inserted. The height of the boiler is, of course, important, but it is a variable that is fixed by other conditions in naval service, such as the deck heights and draft of the ship, etc. An increase in the amount of radiant surface would reduce the heat input per square foot, but this is usually determined by the space available, and it will be well to assume that this also is fixed by outside circumstances. The use of large tubes near the fire as a method of relieving the load on fire row tubes will not be considered, as it is assumed that this is well known and much used, and need not be considered in the present case.

The theory which has been used in this paper is based on the following assumptions:

1. Self vaporization in upcomers is nil.
2. Retardation of vaporization in downcomers is nil.
3. Upcomer tubes do not discharge into solid water in the steam drum, but into a light froth the density of which is sufficiently near that of the mixture in the tubes so that its effects may be ignored.
4. Any separation occurring in the water drums will not affect the differential pressure in those drums, i.e., no separation can be assumed for calculating the differential pressure.
5. Flow is sufficiently turbulent in tubes to insure that mixture of water and steam is homogeneous, i.e., that the steam bubbles do not rise relative to the water.

In making these calculations the analysis proceeded along the following lines:

1. Temperature gradient curve for flue gas was computed.
2. Pounds of steam generated per tube row was computed from (1).

3. According to Bernoulli's theory of hydraulic flow, certain definite relationships exist between velocity, average density in the tube row and the steam generated in the tube row.

4. Mass flowing down through the downcomers must equal the mass flowing out of the upcomers.

5. Number of downcomers was computed from (3) and (4).

6. Differential pressure in the water drum was computed from (3) and (4).

The results of the theoretical study showed remarkably good checks with the following observed test data:

1. Temperature gradient curve.
2. Steam generated.
3. Differential pressure in water drums at various loadings.
4. Division of tube rows into downcomers and upcomers from results obtained with pitot tubes.

Throughout this discussion adequate circulation is defined as that flow in the fire row tubes necessary to provide sufficiently good heat transfer coefficients from tube wall to water so that the tube wall will not be damaged by overheating. This, of course, is a function of the mass velocity and the spouting density for these tubes.

The general results from this analysis are presented in the following paragraphs. Even if the actual quantitative values may be questioned because of assumptions made, or simply because of the complexity of the calculation itself, still the results may show trends which can be used qualitatively to advantage.

The development of Bernoulli's theory for hydraulic flow as applied to the A-type boiler has been fully indicated in Appendix A. In Appendix B are sample calculations showing the method used in estimating circulation in the White-Forster boiler. In Appendix C are curve sheets which represent the equations derived in Appendix A. Appendix D contains curve sheets indicating the circulation in the typical A-type boiler. The results indicated on these curves were computed, by means of the theory, for the White-Forster boiler which Admiral Robinson referred to in his paper on "Water Circulation and Gas Path in Naval Boilers." The design details referred to in the discussion and summary have been placed in Appendix E.

Separation in the Steam Drum.

The separation of the steam from the water is the chief function of the steam drum. Usually the measure of the efficiency of the steam drum in performing its function is the percentage of moisture leaving the dry pipe; the less moisture the better the design. With boilers that are highly rated there is another criterion that is of equal importance. The separation of the steam must not only be performed without undue carry-over of water, but it also must be complete. That is, all the steam delivered by the upcomers must be separated from the water in the steam drum. If this is not accomplished the downcomers will receive a mixture of water and steam instead of solid water. The boiler in this case will circulate a quantity of steam through its tubes in a repetitive cycle which it never delivers to the steam space. This general lowering of the density level is in itself serious, but, in addition, there is also a reduction of the mass velocities which can be induced in the fire row tubes.

Before considering in detail the magnitude of the effects of lack of separation in the steam drum, it will be well to determine, if possible, the probability of steam entering the downcomers before having opportunity to free itself and pass into the steam space. In marine A-type boilers in this country it has been the general practice to install a flat Vee baffle below the water level in the steam drum about eight to twelve inches above the point where the fire row tubes enter the drum, as shown in Appendix E, Figure 1. This baffle is so designed that the blast of steam and water, issuing from those tube rows nearest the fire, impinges on the baffle and is deflected to the sides of the steam drum. This prevents violent ebullition from carrying particles of water to the dry pipe with a consequent increase in the percentage of moisture in the steam. Since, however, the point toward which the steam and water mixture is deflected, is the point at which the downcomer tubes take suction, it would appear highly probable that a large amount of steam would be carried down the downcomer tubes.

In my opinion there is no better way of improving this condition than by installing a circulation augmentor as described by Engineer-Captain S. R. Dight, C.B.E., R.N., in his paper on "Naval Water Tube Boilers," published in "Transactions of Institution of

Naval Architects" for 1936. The tube rows nearest the fire, which deliver the greatest quantities of steam, discharge into a trough which separates it from the remaining water in the steam drum as shown in Appendix E, Figure 2. As the steam-and-water mixture flows over the top of the trough there is ample opportunity for the steam bubbles to free themselves. The feed pipes and other internal fittings are placed outside of the trough in the space just above the point where the downcomers are attached to the drum. This arrangement has an additional advantage in that the trough at high rates of evaporation is always filled with a low-density mixture which offers a minimum of resistance to the discharge of water from the upcomers. The Vee baffle offers a great deal of resistance to this discharge, since it prevents the formation of a region of low density, which the upcomers naturally tend to build up in this section of the drum, even without the help of the circulation augmentor. This circulation augmentor increased the rate of firing of a given boiler by about 20 per cent, according to Captain Dight's paper. This is additional indication that separation in the steam drum is a large factor in obtaining adequate water circulation.

In Appendix D, Figure 4, the water circulation in the fire row tubes as computed by the theory described has been plotted against the variation in the density of the water entering the downcomer tubes. This shows that, if the upcomers receive the mixture which the downcomers deliver, the mass velocity and spouting density will both vary linearly with the variation in density to downcomers. If, however, the fire row tubes receive solid water due to separation occurring in the water drums (this is discussed in detail in the section following) then the reduction in mass velocity and in the spouting density will be much larger for a given decrease in density to the downcomers. It is this latter factor which makes it so extremely important to obtain complete separation in the steam drum. It is, in effect, a pyramiding of conditions that tend to prevent good water circulation.

Separation in Water Drums.

Separation of steam bubbles from the mixture of steam and water passing through the water drums, from downcomers to upcomers, need be considered only in instances when the downcomers

are heated sufficiently to cause the formation of steam. Even in these instances, this possibility of separation is often ignored and it is assumed that all upcomers receive the mixture which the downcomers have brought them. Such an assumption is not tenable. The steam drum, where, of course, separation does occur, is frequently filled with internal fittings and baffles. These baffles are not present in the water drum, so that in these drums there is an even greater opportunity for separation. Therefore, it must be expected that in the water drums separation will occur.

At the high ratings at which A-type boilers operate for naval service the density of this mixture from the downcomers is very low and the effects of separation cannot be disregarded. Any separation occurring in this drum will have large effects on the fire row tubes. These tubes usually leave the water drums at a point lower than do any others and, since any separation is due to the tendency of the steam bubbles to rise against gravity or to float due to their low density, it naturally follows that the fire row tubes will receive water with a lower quantity of entrained steam than will any other tube row. This condition has an extremely detrimental effect on the circulation in these tubes.

For the purpose of better understanding the reason for this, let us consider it from a physical point of view with a minimum of mathematics. Let us assume that the fire row tubes receive solid water. Then the weight of the mixture in this upcomer will be increased and the pressure in the water drum will not impart as high a mass velocity to this mixture as would have been the case had this upcomer received its share of the steam bubbles delivered by the downcomers. In addition, the spouting density of this tube row will not be improved by receiving solid water. The quantity of the steam formed in a given tube row is determined by gas flow conditions * and is independent of water circulation; but the quantity of steam that is delivered is a function of mass velocity and the difference in the densities of the mixture entering and leaving the tube row. Now the density entering has been improved, but the detrimental effects on mass velocity over-balance this gain and in most instances, in order that the tube row may deliver the

* The heat transfer coefficient is a function of gas velocity only because the resistance to heat flow on the water side is very slight compared with that on the gas side.

requisite steam quantity, the spouting densities must be even poorer than would have been the case if the tube row had received its share of steam bubbles.

Numerous tests on the water circulation in A-type boilers have contributed data which corroborate the physical concept of separation in the water drums given above. In 1933 Rear Admiral S. M. Robinson, U. S. N., presented a paper before the Society of Naval Architects and Marine Engineers on "Water Circulation and Gas Path in Naval Boilers." In this paper are the results of tests on the White-Forster boiler, which I have used as a representative A-type boiler, and on a Babcock & Wilcox Express (A-type) boiler. There are three points in particular that show experimentally the possibility of separation occurring in the water drums.

The first is the reading of the gauge glass on the water drums. This glass indicates that the density in the water drums is not that of solid water at ratings where downcomers are heated sufficiently to cause the formation of steam. This observation has led to the conclusion that there is a steam pocket in the top of the water drum and this is, I believe, correct, although no doubt it is a very small pocket, since the glass would show a low level even if the mixture in the drum were homogeneous. The second point is the observations made from pitot tubes. In both A-type boilers investigated, a strong upcomer was discovered, in a region of downcomers, which was so located in the upper portion of the water drum that it formed a natural vent for the steam bubbles that separated out of the mixture discharged into the water drum by the downcomers. This same venting action was noticeable even in some external downcomers located at the top of the water drums. In every case this venting action occurred only at the higher rates of evaporation. The third point is the pulsating flow that has been observed by numerous investigators. It is doubtful if a homogeneous fluid would behave in this manner and in the tubes, where the flow is in the turbulent range, the fluid must act as a uniform mixture. A logical explanation would appear to be that it is not homogeneous in the water drums, and that variations in the degree of separation occurring there cause the pulsations.

In Appendix D, Figure 5, the circulation in the White-Forster A-type boiler, as computed by the theory developed, has been plotted in such a manner as to show the effects of separation in the water drum on the circulation in the fire row tubes. No external downcomers have been considered and the density of the water to the heated downcomer tubes (i.e., the degree of separation in the steam drum) has been varied from solid water to eight-tenths solid water. On the curves for both mass velocity and spouting density the locus line for no separation in the water drum has been added. These curves show that, if additional steam bubbles could be supplied to the fire row tubes so that the density entering these tubes was about four-tenths that of solid water, the mass velocity and the spouting density would both be at their maximum values. In general, it is apparent, however, that if the fire row tubes merely were to receive their share of the steam bubbles the circulation would be satisfactory.

The danger point, of course, does not depend only on the degree of separation in the water drums. If the downcomers are fed with a mixture of steam and water instead of with solid water, i.e., if the separation in the steam drum is poor, then the effect of separation in the water drums becomes greater. In other words, if the fire row upcomers are receiving solid water and the downcomers are also given solid water the reduction in circulation due to the fire row tubes not receiving their share of steam bubbles is not so great as in the case where the downcomers receive a mixture of steam and water. With solid water entering the downcomers, the mass velocity in the fire row tubes is improved about 40 per cent if the fire row tubes receive the mixture delivered by the downcomers instead of solid water. With a water-and-steam mixture of eight-tenths of solid water entering the downcomers, the improvement would be 300 per cent. The spouting density is improved in about the same ratio.

From this it is evident that the degree of separation occurring in the water drums is of great importance to the water circulation. It may well be that in A-type boilers this physical phenomenon may prove to be the deciding factor between successful and unsuccessful circulation.

External Downcomers.

External downcomers have been used in this country whenever designers have believed, or tests have shown, that circulation was inadequate. The addition of the downcomers has always improved the circulation as required, so that there has been built up a feeling that for high rates of evaporation, external downcomers are a necessity. In England, however, such is not the case. If Captain Dight's paper is a reliable indicator, external downcomers are used only occasionally. It is felt that the additional stiffness between steam and water drums may be detrimental and that the space required is excessive. It seems that their reasoning is sound, since their boilers operate at ratings comparable to ours, despite the fact that we furnish external downcomers and they do not. It is difficult to reconcile these two totally different views, and yet I believe they are reconcilable. In Appendix D, Figure 8, the effect on mass velocity and spouting density of increasing the size of the external downcomers has been plotted. For normal design the ratio of the area for flow through the external downcomers to the area for flow through the tube bank would be about .05. Now if it is assumed that the density entering the downcomers is that of solid water then from these curves it would appear that the increase in mass velocity would be 7 per cent if the fire row tube received solid water and less than 1 per cent if the fire row tube received the mixture delivered by the downcomers.

Since, however, the mass velocity in the external downcomers is large relative to that of the heated downcomers there will be an increase in the turbulence in the water drum, which will hinder separation so that the fire row tubes will more nearly receive their share of steam bubbles. This will improve the velocity in the fire row tube as described in the previous section on "Separation in the Water Drums." In this case, however, the maximum gain theoretically possible is only about 40 per cent. If now the density entering the downcomers is .8 that of solid water, then the increase in mass velocity would be about 40 per cent if the fire row tubes received solid water and about 1 per cent if there were no separation in the water drum. In this case if the external downcomers prevented separation from occurring in the water drums the gain

in mass velocity would be about 300 per cent. Similar effects in spouting density to those mentioned for mass velocity are shown in the figure.

There is one combination of inlet densities to downcomers that should be considered at this time. If the external downcomer receives solid water and the heated downcomers receive a mixture having a density .8 that of solid water, then with a ratio of area of external downcomers to area through tube bank of .11 there will be only approximately one row of heated downcomers and the mixture delivered by all downcomers will be practically solid water. In this case the maximum improvement in mass velocity would be only 200 per cent.

From this, two general conclusions could be drawn: First, if there were no separation in the water drum, external downcomers as used in the past would not improve the water circulation. Second, if the separation in the steam drum were perfectly complete, then the circulation for present ratings would be adequate without external downcomers and the addition of external downcomers would result in only a relatively small improvement.

Depth of Tube Bank.

The effects on water circulation of varying the depth of tube bank (i.e., the number of tube rows having a common water drum) depends on the manner in which the variation in depth is accomplished. Usually a reduction in depth is obtained by omitting some of the tube rows nearest the uptakes. It is also possible, however, to reduce the depth by omitting the tube rows nearest the fire. An example of this type is the Yarrow boiler with divided water drums on one side of the boiler, with the superheater located between the upper and lower water drums. In this case the tube bank for the upper drum has been reduced by omitting tube rows adjacent to the fire. Both of these two types will be discussed.

The effect of reducing the depth of a tube bank by omitting some of the tube rows nearest the fire will be considered first. In this case the danger of burning a tube is very much reduced for two reasons: First, the heat input per tube for a tube which is shielded by merely three rows of tubes is only 25 per cent that of a fire row tube. Secondly, as the depth of the bank is reduced by omitting

tube rows nearest the fire, the density of the spouting mixture is improved and the mass velocity in the tube row nearest the fire in this bank is increased slightly. This improvement in circulation coupled with the drastic decrease in loading on the tube insures that the tubes in this bank nearest the fire will be free from troubles due to inadequate circulation. It therefore seems reasonable to assume that the reduction of depth of tube bank in this manner will never affect the circulation adversely.

The method of reducing the depth of tube bank by omitting tube rows adjacent to the uptakes has not so fortunate an effect on circulation. If there were no separation in the water drums, circulation would be excellent, but, as previously discussed, this assumption can scarcely be accepted. If, as is probably the case, complete separation for fire row tubes is assumed in the water drums, then the mass velocity in these tubes diminishes rapidly as the depth of tube bank decreases. For a tube bank only twelve to sixteen tubes deep, circulation is definitely inadequate for high ratings. From Appendix D, Figure 6, it is evident that the degree of separation in the steam drum determines the depth of tube bank that will fail to provide the necessary water circulation without external downcomers or some other provision for insuring no separation in the water drums. For complete separation in the steam drum a depth of eighteen tube rows might be sufficient at this rate of evaporation, but if the density entering the downcomers is only .8 that of solid water then even twenty-four rows of tubes are insufficient.

At this time a distinction should be made between a depth of tube bank of, say, twenty tubes and one of six tubes. In the former case there would be approximately eleven upcomers and an assumption that the fire row tubes fail to receive their share of steam bubbles is perfectly reasonable. In the latter case only three tube rows are upcomers and it is inconceivable in such an instance that the distribution of steam bubbles to these fire row tubes would be poor, since there is so little distance between downcomer discharge and upcomer suction and since, in most cases, these upcomers are located practically at the top of the water drum. That this point of view is entirely justified is shown by the successful operation of the Yarrow boiler with divided water drums mentioned above. In

this case there is a five- or six-row water screen forming one complete tube bank between the superheater and the fire. This design has been steamed at ratings comparable to those used in naval service without the use of external downcomers. This physical concept of separation in the water drums offers a most acceptable theory for explaining why this tube bank with such a shallow depth maintained satisfactory circulation at evaporation rates at which tube banks of more normal depths failed to do so.

From this discussion it seems that the effect on water circulation of varying the depth of tube bank is dependent upon the degree of separation occurring in the water drums. In fact, it can be said that the circulation effects, which have been observed from tests on boilers of various depths of tube banks as with similar observation with and without downcomers, have been explained by this concept of separation in the water drums and that no other physical concept shows an adequate explanation of either of these.

Summary.

In any A-type boiler operated at high rates of evaporation, there are two distinct groups of factors which influence the water circulation occurring in the fire row tubes. The first group is composed of geometric considerations such as the boiler height, the caliber length of the tubes, the angle at which the tube bank is inclined, and the size of the furnace available to burn the quantity of oil required. These factors are determined very early in the design of a boiler and usually are fixed by the service demanded and the space available. Frequently the design obtained on this basis does not provide adequate circulation. A change in this first group of factors influencing the circulation simply avoids the problem by changing the design conditions and cannot be considered as a satisfactory solution. It is to the second group that we should look for methods of obtaining adequate water circulation without sacrificing any of the original requirements.

This second group is composed of four factors which influence circulation:

1. Separation in steam drums.
2. Separation in water drums.
3. External downcomers.
4. Depth of tube bank.

From our previous discussion of these variables, it has been found that the last two are not independent variables, but are simply factors which nullify or correct the effects of the first two. For the maximum mass velocity and spouting density in the fire row tubes the following flow pattern is necessary. In the steam drum separation from the water of the steam bubbles delivered by the upcomers should be complete so that the downcomers will receive solid water. In the water drums, separation of the steam bubbles from the mixture brought down by the heated downcomers must not occur. These bubbles must remain in supsension so as to supply a homogeneous fluid to the fire row upcomer tubes. If only one or the other of these two flow conditions is departed from, the detrimental effects on circulation will not be large; but, if neither flow pattern can be maintained, there will be serious reductions in the circulation in fire row tubes.

Separation in these drums can be controlled only by varying the turbulence in these drums or by changing the direction of flow through the drums. It is the usual practice in this country in A-type boilers to install baffles and internal fittings below the water level in the steam drum which increases the turbulence in this drum. This increase in turbulence tends to mix the steam and water and hinders instead of aids the separation. In England the circulation augmentor, recommended by Captain Dight, which reduces turbulence in the steam drum and gives better flow direction to the steam bubbles delivered by the upcomers, has been used on several occasions with gratifying results. This is, I believe, sound design and we would do well to follow it.

The flow in the water drums should also be altered to improve the circulation in the boiler. In existing design these drums have not been fitted with any internal encumbrances to flow, so that the turbulence in these drums has been at a minimum. The turbulence should be increased by the addition of baffles which would maintain velocities throughout the drum and which would prevent the flow from downcomer to upcomer from picking up solid water from the lower portions of the drum. This turbulence would prevent separation of steam and water and would insure that the fire row tubes received their share of the steam bubbles delivered to the water drums by the heated downcomers. It is only by maintaining a low

density in these fire row tubes by supplying them with a mixture of steam and water that adequate water circulation can be obtained.

In the past, external downcomers or a large depth of tube bank have been considered essential for good water circulation at high rates of evaporation. The development of this theory has indicated that these have been necessary because of the separation which occurred in the water drums. This is the only condition where external downcomers of small size relative to the size of the tube bank improve the water circulation in the fire row tubes. Even if the internal downcomer tubes delivered large quantities of steam to the water drum, small size external downcomers would not improve the circulation in the fire row tubes unless these tubes did not receive their share of the steam bubbles. This is also true with regard to the necessity of using a large depth of tube bank for boilers designed for high ratings.

To complete this summary, two A-type boiler designs have been developed in which an attempt has been made to embody the results obtained from the theory that has just been presented. The first of these sketches is shown in Appendix E, Figure 3. This boiler should prove adequate for the present loadings to which naval boilers are subjected. In this design, the flow in the steam and water drums has been suitably baffled to diminish turbulence in the former drum and to increase it in the latter. These changes should provide better flow conditions in the drums and should improve water circulation in the fire row tubes by a large amount, since the flow conditions which improve circulation rather than those which do not will be pyramided. In this instance, external downcomers would never be used except when the depth of tube bank was so shallow that the flow area through the external downcomers could be about 50 per cent of the flow area through the tube bank which they serve, without the size of the external downcomers becoming so excessively large as to be impracticable.

The second design is a more complete development of the theory. Everything which the theory indicated would tend to accelerate circulation has been incorporated. In this design each tube bank has been divided into two tube bundles by a partition in the water drum. This partition separates the first six tube rows nearest the fire from the remaining tubes. In effect, the original tube bank has

been separated into a highly loaded tube bank and a normally loaded one. The tube bundle nearest the fire is equipped with external downcomers having a flow area equal to at least 50 per cent of the flow area through these tubes. Since these downcomers serve only six tube rows, their size will not be prohibitive. In fact, their size will be about the same as those that have been fitted in the past on A-type boilers. External downcomers of this size would probably increase the circulation in these six tube rows about 50 per cent and might even double it at a given rate of evaporation. The remaining tube bundle is not fitted with external downcomers, but the drums are fitted with baffles similar to those used in the first design. This should, of course, be perfectly adequate, for the maximum heat input to any tube in their tube bundle would be low compared to that for a fire row tube. Of course, the pressure in the section of the water drum serving the first six rows nearest the fire will be higher than in that section serving the other rows, so that the partition must be tight to avoid leakage. This pressure differential is very low, however, probably only about one or two feet of water as a maximum, so that the strength of the baffle is not a problem. A sketch of this design is shown in Appendix E, Figure 4.

Conclusions.

This paper has presented a theory which attempts to show the water flow conditions existing in an A-type boiler. The work of developing this theory was undertaken in the hope that a more complete knowledge of the fundamentals controlling the water circulation would indicate whether existing boiler designs have made complete use of all the forces producing natural circulation. If these existing designs have failed to do this, then the circulation in the fire row tubes has not approached the maximum theoretically possible; and the heat input which these tubes can absorb has been limited, unnecessarily, by improper design. This is, of course, of the utmost importance, since the size and weight of a naval boiler depend directly upon the ability of these tubes to absorb the heat delivered by the flue gas.

In the last ten years difficulty has been experienced in obtaining sufficient improvements in water circulation to keep pace with the

increase in capacity of furnaces and burners. Gains in circulation, which have sufficed for the present, have been obtained by using larger external downcomers whenever an increased rate of evaporation was desired. The development of this theory has indicated that these existing designs have not obtained the maximum heat absorbing ability possible with natural circulation in an A-type boiler. The conditions which would produce inadequate circulation if external downcomers were omitted have not been altered; the external downcomers have provided adequate circulation in spite of these conditions and at the expense of size and weight in the boiler design. The flow area through these external downcomers as fitted at present is only about 5 per cent to 12 per cent as a maximum of the flow area through the tube bank which they serve. To obtain the maximum circulation theoretically possible with natural circulation by simply increasing their size would require that their flow areas be increased to 50 per cent of the tube bank area. The cost in weight and space to do this would be prohibitive.

The use of forced circulation boilers has been suggested as the next step in the development of the marine boiler for high capacity service. In my opinion such a step should not be made until the natural circulation boiler has been developed to the point where the absorbtion of greater heat inputs is inherently impossible. The operating risks of forced circulation are large and should not be accepted unless it is definitely certain that the increase in capacity can be obtained in no other way.

If the natural circulation A-type boiler, capable of delivering a given quantity of steam, is to be comparable in weight and space to a forced circulation boiler, the trend in design for the future must make an abrupt departure from that of the last ten years. A more economical use of the fundamental factors producing the circulation must be obtained. It is only thus that the size of external downcomers can be reduced so that the weight and space occupied by a boiler will depend more on the amount of heating surface and on the efficiency desired than upon the provisions for obtaining adequate circulation.

The purpose of any theoretical paper of this sort is not to establish such a design, but to show the direction along which developments should proceed. As a means of performing such a function,

there are included in the summary two designs based on the theory which attempt to indicate possible means of attaining the three conditions which would be the most beneficial to the circulation. These are separation of steam from the water in the steam drums, no separation of steam bubbles from the mixture of steam and water in the water drums, and the use of external downcomers for the first few tube rows nearest the fire. From theoretical considerations, the improvement in circulation due to these factors appears to be very large, which, of course, means that the heat input may be increased considerably. Some experimental observations have been cited in the previous sections which have in some small part corroborated portions of the theory.

The experiments conducted in England on the circulation augmentor would seem to indicate that large increases in boiler ratings may be expected if the conditions for separation in the steam drum are improved. In view of the apparent gains and in view of the fact that previous experimental work verifies in part that these gains are possible of attainment, it seems that there is sufficient justification to warrant some additional experimental verification of the theory. Since the two designs submitted are merely modifications of the usual A-type boiler, the actual test work should not be expensive.

It has been felt that in order to develop the A-type boiler for the high rates of evaporation for which it is inherently capable, development must proceed by basing experimental research on theoretical considerations. In view of this belief, this theory of circulation is presented in the hope that it will lead to a clearer conception of the fundamental factors which govern the flow through the tubes. It is also hoped that this work will contribute in some small part to the future development of high-capacity marine boilers.

APPENDIX A.

Derivation of Equations of Flow.

NOMENCLATURE.

v_1 = Inlet linear velocity of mixture in tube in feet per second.

v_o = Outlet linear velocity of mixture in tube in feet per second.

$\overline{V}$ = Mass velocity of mixture in tube in pounds per square foot per second.

Q = Steam generated in tube in pounds per square foot per second.

d_w = Density of solid water at saturation temperature in pounds per cubic foot.

d_s = Density of steam at saturation temperature in pounds per cubic foot.

d_1 = Inlet density of mixture in tube in pounds per cubic foot.

d_o = Outlet density of mixture in tube in pounds per cubic foot.

d_a = Average density of mixture in tube in pounds per cubic foot.

x_1 = Inlet percentage of volume occupied by steam.

x_o = Outlet percentage of volume occupied by steam.

C_1 = Number of velocity heads to be supplied.

C_2 = Number of velocity heads for friction through tubes.

fl/D = Friction group ($1/D$ = length, diameter ratio of tube; f = friction factor).

h = Pressure in water drum above steam drum pressure in feet of water at saturation temperature.

H = Height of water level in steam drum above lower lip of tubes in water drum.

CONSIDER FIRST AN UPCOMER TUBE.

Set up Bernoulli's equation between a section just after the inlet, and a section just before the outlet of the tube.

Then :

$$\frac{d_o v_o^2}{2g} + Hd_s = hd_w - .5d_1 \frac{v_1^2}{2g} + \frac{d_1 v_1^2}{2g} - d_a \frac{fl}{D} \frac{v_a^2}{2g}$$

where, $hd_w - .5d_1 \frac{v_1^2}{2g}$ = pressure at section just after the inlet of the tube.

simplifying,

$$hd_w - Hd_a = \frac{\overline{V}^2}{2g} \times \frac{1}{d_a}\left[\frac{d_a}{d_0} - \frac{.5d_a}{d_1} + C_2\right] = \frac{\overline{V}^2}{2g} \times \frac{C_1}{d_a}$$

where,

$$C_1 = \frac{d_a}{d_0} - \frac{.5d_a}{d_1} + C_2$$

FOR THIS ANALYSIS ASSUME:

$$d_a = \frac{d_0 + d_1}{2}$$

Now $\left(\frac{1}{4} - \frac{d_0}{4d_1}\right)$ is infinitesimal compared to $\left(\frac{d_1}{2d_0}\right)$ and compared to usual values of C_2

Therefore:

$$C_1 = \frac{d_1}{2d_0} + C_2$$

SOLVING FOR $\overline{V}$:

$$\overline{V} = \frac{d_1\sqrt{2gH}\sqrt{\left(\frac{b}{a}+1\right)\frac{1}{2}\left[\frac{h}{aH} - \left(\frac{b}{a}+1\right)\frac{1}{2}\right]}}{\sqrt{\frac{a}{2b} + C_2}}$$

where

$$a = d_1/d_w$$
$$b = d_0/d_w$$

IN LIKE MANNER THE EQUATION FOR DOWNCOMERS MAY BE WRITTEN AS:

$$\overline{V} = \frac{d_1\sqrt{2gH}\sqrt{\left(\frac{b}{a}+1\right)\frac{1}{2}\left[\left(\frac{b}{a}+1\right)\frac{1}{2} - \frac{h}{aH}\right]}}{\sqrt{\frac{a}{2b} + C_2}}$$

OR IN MORE GENERAL FORM:

$$\overline{V} = d_1 \sqrt{2gH} \qquad f(V)$$

DETERMINATION OF STEAM GENERATED IN A HEATED TUBE:

Let A = Internal area of the tube in sq. ft.
Steam entering tube,

$$\text{Wt of steam in lbs./sec.} = \overline{V}\, A\, x_1\, d_s/d_1$$

$$\text{Wt of steam in lbs./sec.} = \frac{\overline{V}\, A\, d_s\, (d_w - d_1)}{(d_w - d_s)\, d_1}$$

In like manner, steam leaving tube,

$$\text{Wt of steam in lbs./sec.} = \frac{\overline{V}\, A\, d_s\, (d_w - d_0)}{(d_w - d_s)\, d_0}$$

THEREFORE, STEAM GENERATED IN ANY TUBE IN POUNDS PER SQUARE FOOT OF CROSS SECTIONAL AREA PER SECOND MAY BE WRITTEN AS:

$$Q = \overline{V}\, d_s \left[\frac{d_w - d_0}{(d_w - d_s)\, d_0} - \frac{d_w - d_1}{(d_w - d_s)\, d_1} \right]$$

Now since d_s is small relative to d_w, $(1 - d_s/d_w)$ may be assumed to equal (1).
Therefore:

$$Q = \overline{V}\, \frac{d_s}{d_1} \left(\frac{a}{b} - 1 \right)$$

Or in more general form:

$$Q = d_s \sqrt{2gH} \qquad f(Q)$$

APPENDIX B.

DETERMINATION OF C_2 VALUE.

$C_2 = \frac{f1}{D}$. Now assume 1(1) = 120". D = .810"

From Page 257 of "Marine Steam" published by Babcox & Wilcox.

$$f = k\,(1 + 36/10\,D)$$

Now for Steam $k = .0027$

For Water $k = .005$

Using an average value of k

$$C_2 = 3.0$$

TABLE OF PARTICULARS OF THE WHITE-FORSTER A-TYPE BOILER.

Tube diameter in inches	1.000	
Tube thickness	.109	
Tube length (average) in feet	7.16	
Number of tubes	4032	
Number of tube rows (not including superheater)	24	
Number of tubes per row	84	
Boiler water heating surface in square feet (B. W. H. S.)	7565	
Radiant surface in square feet	160	
Furnace volume in cubic feet	751	
H (Hydraulic head in feet)	7.92	
Boiler pressure in pounds per square inch absolute	300	
Density of steam in pounds per cubic foot	.65	
Density of water in pounds per cubic foot	52.8	
Steam rates in pounds per hour	108,000	153,467
Gas rate in pounds per hour	147,000	199,000
Oil rate in pounds per square foot of B. W. H. S.	1.0	1.5
Oil rate in pounds per square foot of radiant surface	47.3	70.9

DETERMINATION OF DIFFERENTIAL PRESSURE IN WATER DRUMS OF WHITE-FORSTER BOILER.

1.5 pounds of oil per square foot of B. W. H. S.

$H = 7.02$ feet. $d_w = 52.8$ pounds per cubic foot. $d_s = .65$ pound per cubic foot.

$Q = 14.65\, f(Q)$. $\overline{V} = d_1\, 22.55\, f(V)$. $d_1/d_w = .90$.

Tube Nos.	① No. of Rows	Pounds of Steam per Square Feet per Second	f(Q)	f(V)	$\frac{b}{a}$	d_o	d_1	$\overline{V}$	$\overline{V}$①	Total	①Vd_o	Total	$\frac{h}{aH}$	h
1	1	22.75	1.55	.160	.095	3.48	35.6	128	128				.933	5.00
2	1	13.37	.912	.184				148	148					
3	1	6.92	.471	.201				162	162					
4-12	9	4.41	.301	.198				159	1433					
17	1	1.68	.1145	.168				135	135					
18	1	1.505	.1027	.163				131	131	2137				
20-28	10	.988	.0673	.201	.75	35.6	47.5	216	2160	2150			.700	

Determination of Differential Pressure in Water Drums of White-Forster Boiler.

1 pound of oil per square foot of B. W. H. S.

Tube Row Nos.	① No. of Rows	Pounds of Steam per Square Feet per Second	f(Q)	f(V)	$\frac{b}{a}$	d_o	d_1	$\overline{V}$	V①	Total	①$\overline{V}d_o$	Total	$\frac{h}{aH}$	h
1	1	14.85	1.012	.212	.175	6.72	38.5	184	184				1.04	5.43
2	1	9.42	.643	.232				201	201					
3	1	5.23	.357	.239				208	208					
4–11	8	3.21	.219	.232				202	1618					
12	1	1.612	.110	.215				187	187	2398				
17	1	.777	.053	.160	.74	35.1	47.5	171	171		6,000		.760	
18	1	.726	.050	.170	.76	36.1		182	182		6,600			
19–28	10	.571	.039	.194	.82	38.9		207	2070	2423	80,600	93,200		

Determination of Possible Mass Velocities in Fire Row Tubes for White-Forster Boiler.

$H = 7.92$ feet. $d_w = 52.8$ pounds per cubic foot.

$Q = 14.65\, f(Q)$. $V = d_1\, 22.75\, f(V)$.

$d_s = .65$ pound per cubic foot.

Oil rate, pounds per square foot of B. W. H. S. ..... 1½ pounds

Pounds of steam per square foot per second	22.75				
d_1/d_w to downcomers = a	1.00				
d_o/d_w from downcomers = b	.75				
h	5.55				
d_1/d_w to upcomers = a	.0123	.35	.60	.80	1.00
h/aH	57.0	2.00	1.17	.875	.700
f(Q)	Supht.	1.55	1.55	1.55	1.55
f(V)	4.00	.394	.220	.141	.086
$\overline{V}$ = Mass Velocity	58.6	164	157	134	102
d_o	.65	3.77	3.89	3.51	2.74

DETERMINATION OF POSSIBLE MASS VELOCITIES IN FIRE ROW TUBES FOR WHITE-FORSTER BOILER.

$H = 7.92$ feet. $d_w = 52.8$ pounds per cubic foot.

$Q = 14.65\, f(Q)$. $V = d_1\, 22.75\, f(V)$.

$d_s = .65$ pound per cubic foot.

Oil rate, pounds per square foot of B. W. H. S. ..... 1½ pounds

Pounds of steam per square foot per second 22.75

d_1/d_w to downcomers = a	.90				
d_o/d_w from downcomers = b	.675				
h	5.00				
d_1/d_w to upcomers = a	.0123	.315	.60	.80	1.00
h/aH	50.3	2.00	1.05	.789	.631
f(Q)	Supht.	1.55	1.55	1.55	1.55
f(V)	3.79	.394	.190	.115	.057
$\overline{V}$ = Mass Velocity	55.5	148	136	109	68
d_o	.65	3.41	3.48	2.83	2.06

DETERMINATION OF POSSIBLE MASS VELOCITIES IN FIRE ROW TUBES FOR WHITE-FORSTER BOILER.

$H = 7.92$ feet. $d_w = 52.8$ pounds per cubic foot.

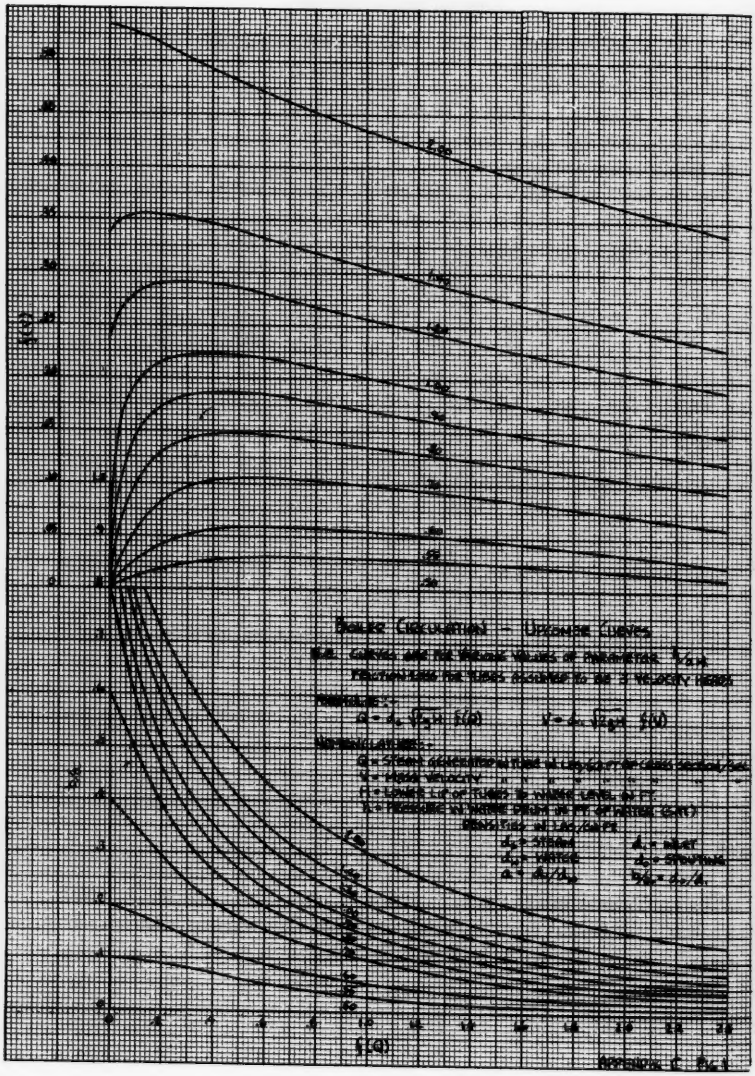
$Q = 14.65\, f(Q)$. $V = d_1\, 22.75\, f(V)$.

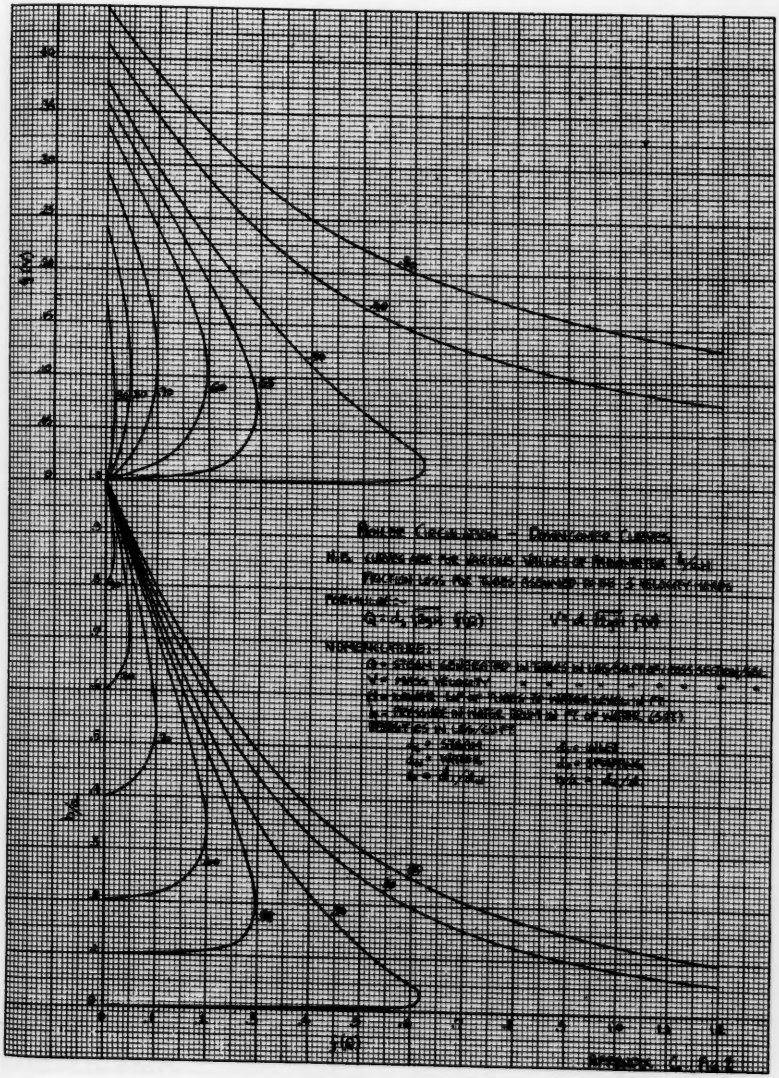
$d_s = .65$ pound per cubic foot.

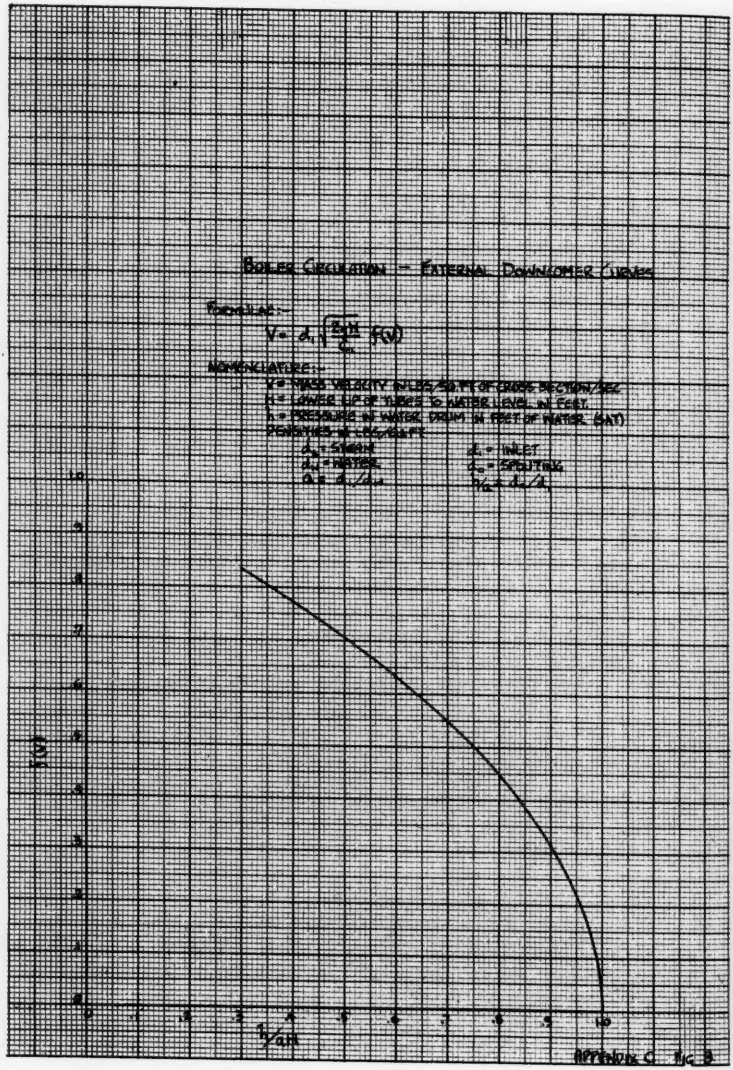
Oil rate, pounds per square foot of B. W. H. S. ..... 1½ pounds

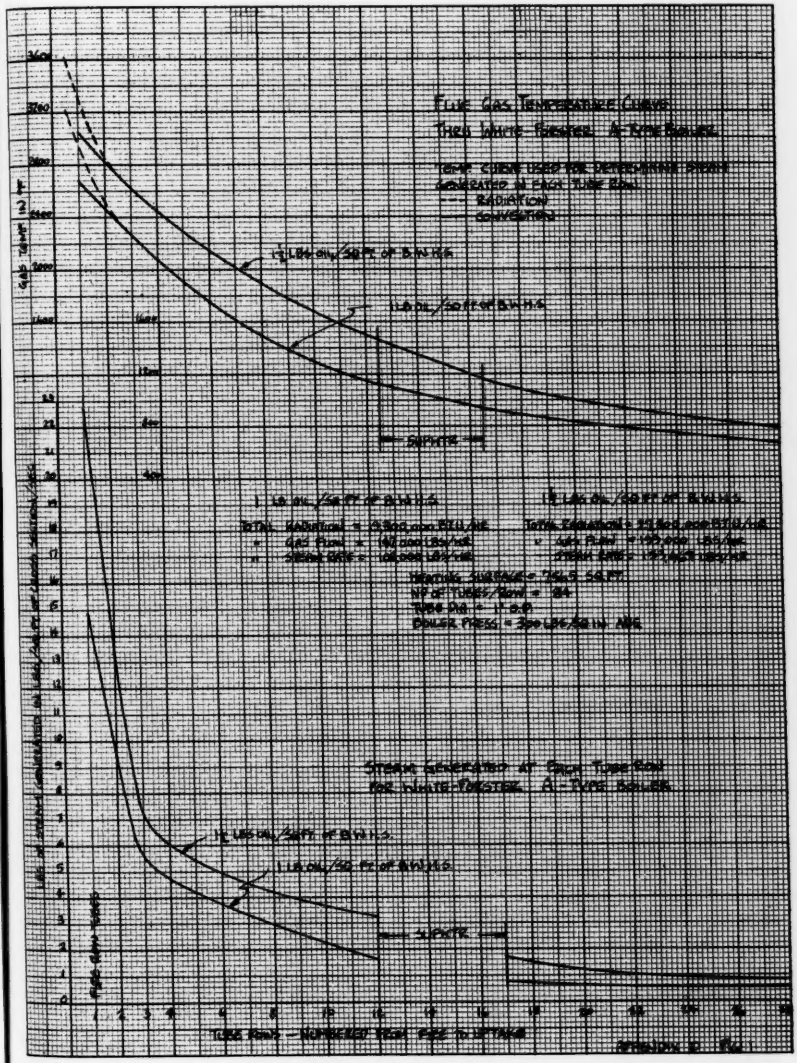
Pounds of steam per square foot per second 22.75

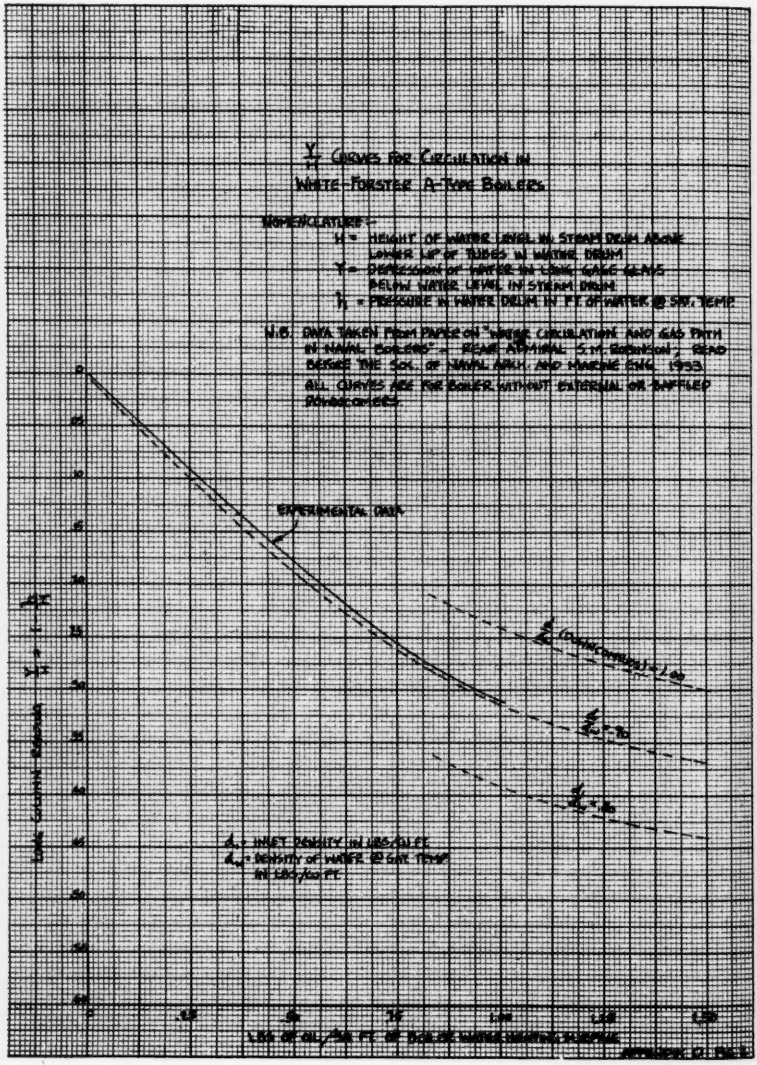
d_1/d_w to downcomers = a	.80				
d_o/d_w from downcomers = b	.60				
h	4.44				
d_1/d_w to upcomers = a	.0123	.28	.60	.80	1.00
h/aH	45.5	2.00	.933	.700	.560
f(Q)	Supht.	1.55	1.55	1.55	1.55
f(V)	3.57	.394	.159	.086	.029
$\overline{V}$ = Mass Velocity	52.3	134	114	82	35
d_o	.65	3.0	2.95	2.20	1.00

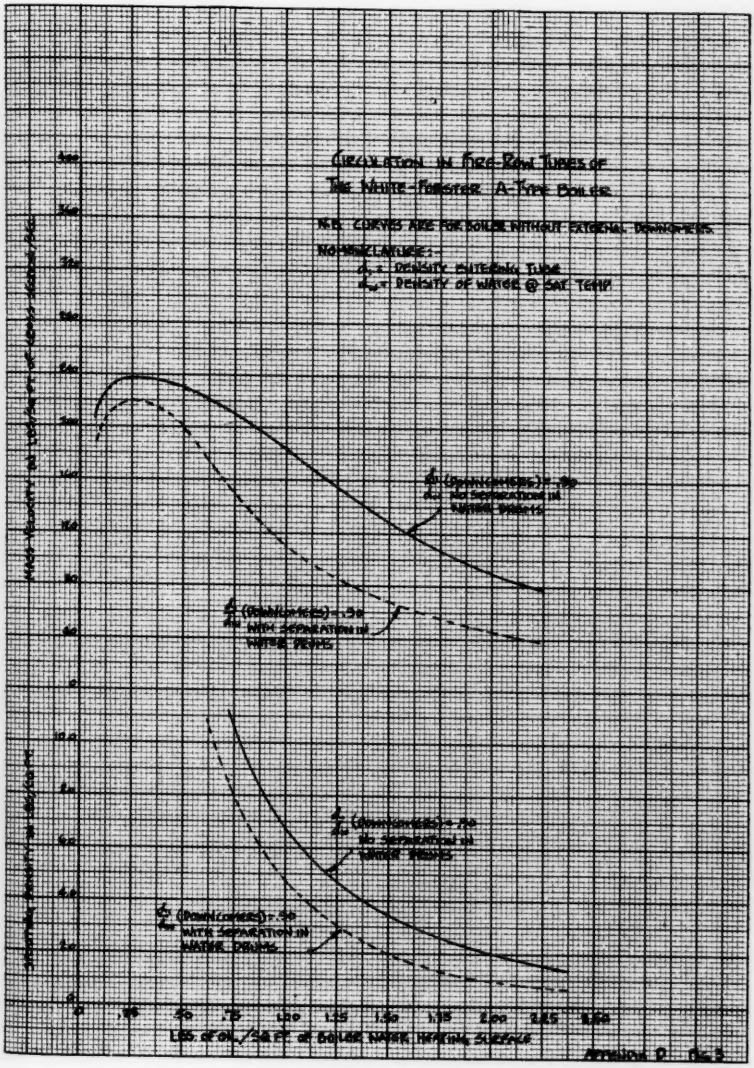

Water Circulation — Discharge Curves
f(Q)

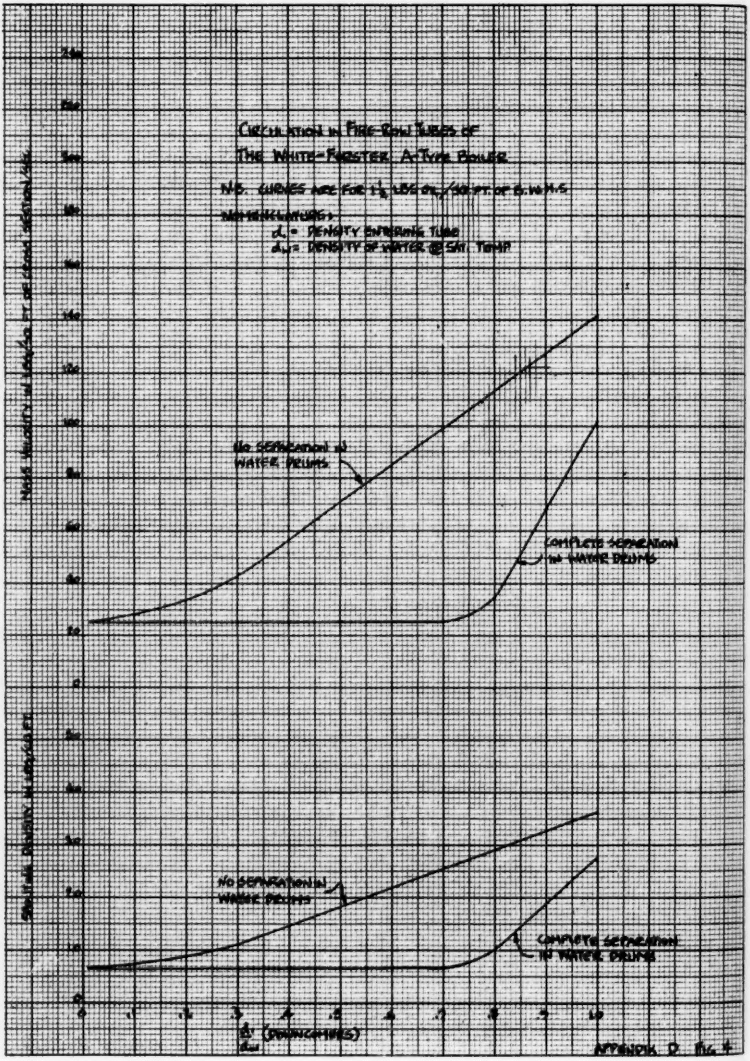


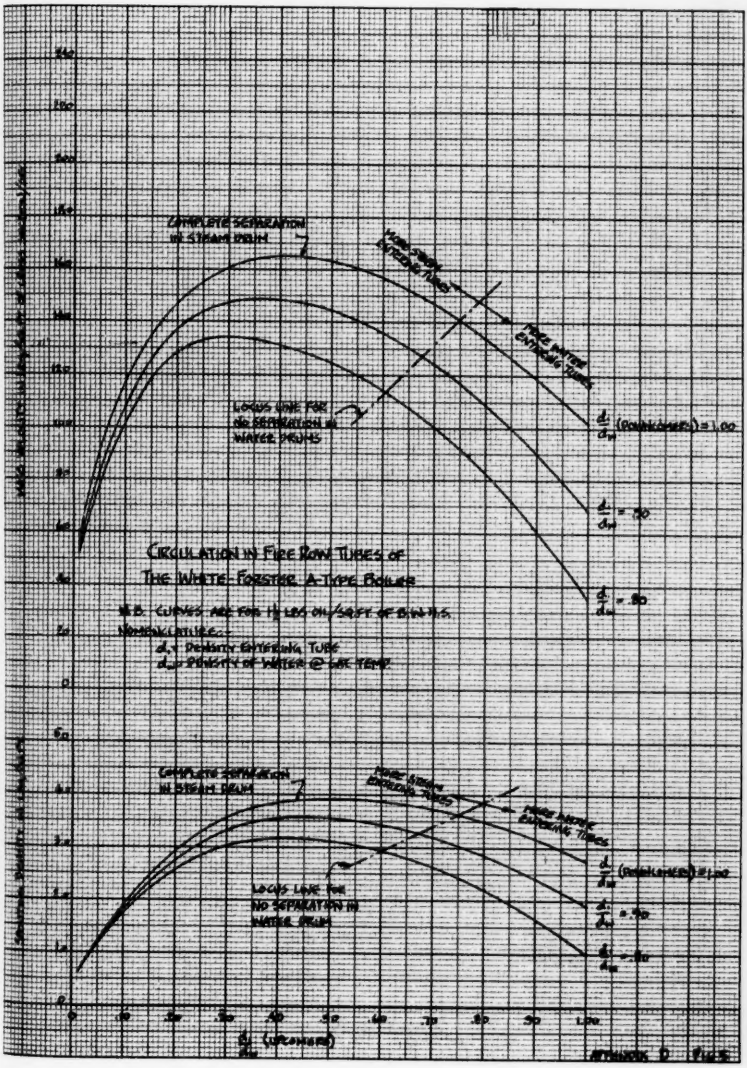

BOILER CIRCULATION — EXTERNAL DOWNCOMER CURVES
FORMULAE:-
NOMENCLATURE:-
V = MASS VELOCITY IN LBS/SQ FT OF CROSS SECTION/SEC
H = LOWER END OF TUBES TO WATER LEVEL IN FEET.
h = PRESSURE IN WATER DRUM IN FEET OF WATER (SAT)
DENSITIES IN LBS/CU FT.
d_s = STEAM
d_w = WATER.
d_i = INLET
d_o = SPOUTING
$r_d = d_o/d_i$
V/dH
1/√dH
APPENDIX C. FIG 3

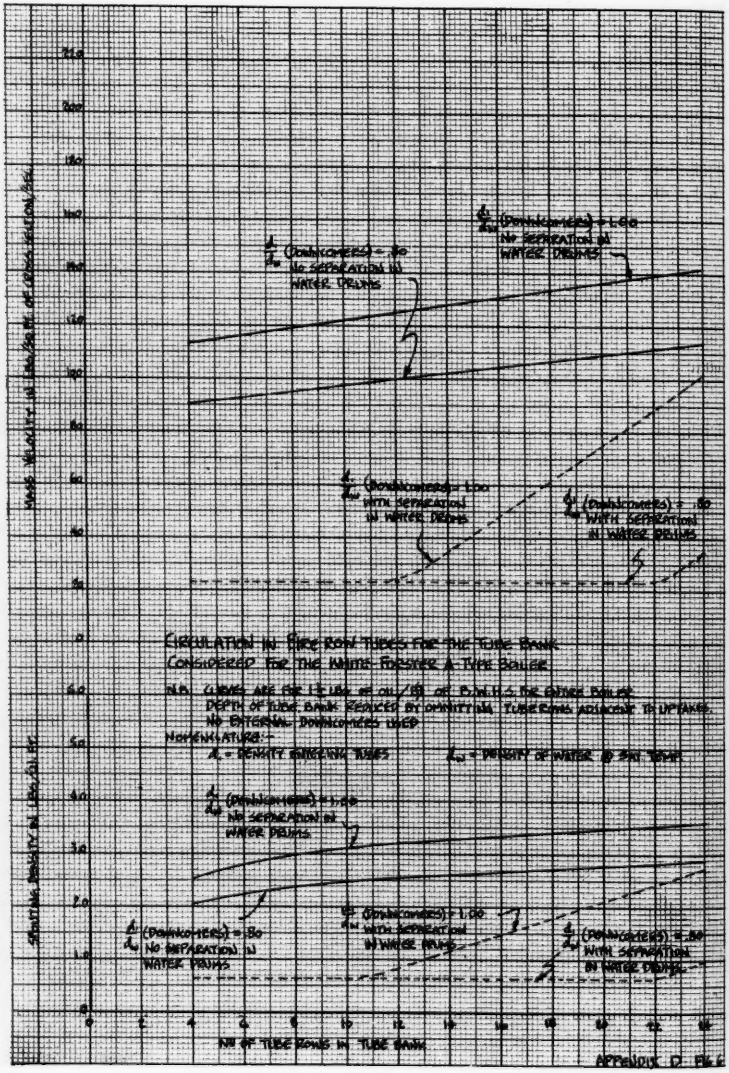

Flue Gas Temperature Curve
Thos. White Forster A-Type Boiler.
Total Heat Absorbed = 9,500,000 B.T.U./hr.
Gas Flow = 140,000 lbs./hr.
Steam Rate = 18,000 lbs./hr.
Heating Surface = 744.9 Sq. Ft.
No. of Tubes/Bank = 84
Tube Dia. = 2" O.D.
Boiler Press. = 300 lbs./sq. in. Abs.
Steam Generated at Each Tube Row
for White-Forster A-Type Boiler.
Tube Row - Numbered from Rear to Uptake

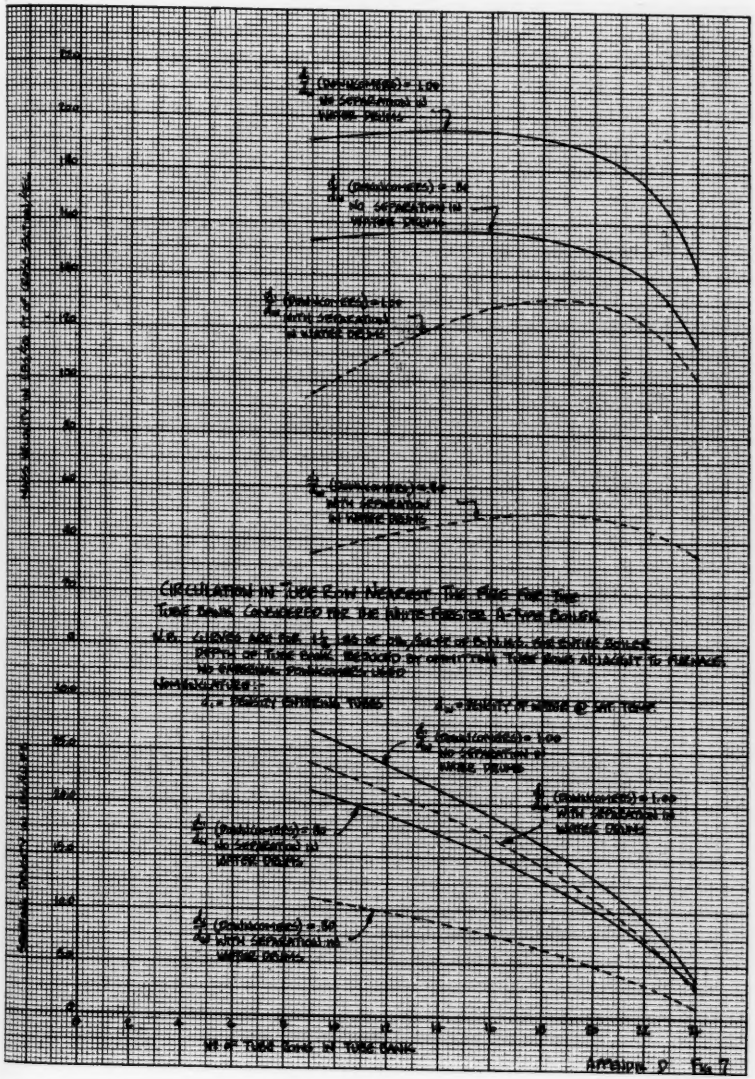

$\frac{V}{H}$ Curves for Circulation in White-Forster A-Type Boilers.
Nomenclature:-
H = Height of water level in steam drum above lower lip of tubes in water drum
Y = Depression of water in the gage glass below water level in steam drum
h_1 = Pressure in water drum in ft. of water @ sat. temp.
U.S. Data taken from paper on "Water Circulation and Gas Path in Naval Boilers" - Rear Admiral S.M. Robinson; read before the Soc. of Naval Arch. and Marine Eng. 1933.
All curves are for boiler without external or baffled downcomers.
Experimental Data
$\frac{V}{H}$ (downcomers) - Lbs
$\frac{V}{H}$ = Lbs
$\frac{V}{H}$ = Lbs
d_1 = Inlet density in Lbs/cu ft.
d_2 = Density of water @ sat. temp in Lbs/cu ft.
Lbs of oil/sq ft of boiler water heating surface

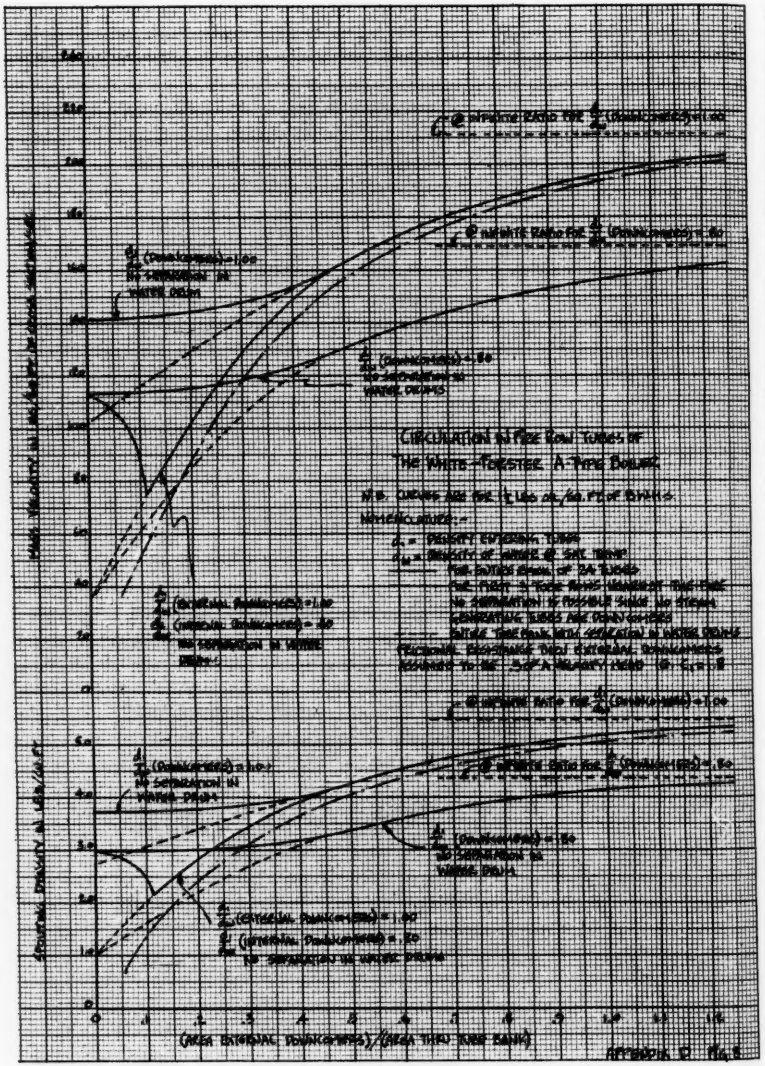

CIRCULATION IN FIRE-ROW TUBES OF
THE WHITE-FORSTER A-TYPE BOILER.
NOTE CURVES ARE FOR BOILER WITHOUT EXTERNAL DOWNCOMERS.
NOMENCLATURE:-
d_s = DENSITY OF STEAM @ DRUM PRESS.
d_w = DENSITY OF WATER @ SAT. TEMP.
$\frac{d_s}{d_w}$ (DOWNCOMERS) = .06
NO SEPARATION IN WATER DRUMS
$\frac{d_s}{d_w}$ (DOWNCOMERS) = .04
WITH SEPARATION IN WATER DRUMS
$\frac{d_s}{d_w}$ (DOWNCOMERS) = .06
NO SEPARATION IN WATER DRUMS
$\frac{d_s}{d_w}$ (DOWNCOMERS) = .04
WITH SEPARATION IN WATER DRUMS
LBS. OF H_2O / SQ. FT. OF BOILER WATER HEATING SURFACE
Approved D. G. S.


CIRCULATION IN FIRE-TUBE TUBES OF
THE WHITE-FORSTER A-TYPE BOILER
N.B. CURVES ARE FOR 1½" I.D.
NO SEPARATION OF WATER DELIVS
COMPLETE SEPARATION OF WATER DELIVS
NO SEPARATION OF WATER DELIVS
COMPLETE SEPARATION OF WATER DELIVS
APPENDIX D FIG 4


COMPLETE SEPARATION IN STEAM DRUM
LOCUS LINE FOR NO SEPARATION IN WATER DRUM
CIRCULATION IN FIRE ROW TUBES OF THE WHITE-FORSTER A-TYPE BOILER
N.B. CURVES ARE FOR HEATING ON LEFT OF BOILER
COMPLETE SEPARATION IN STEAM DRUM
LOCUS LINE FOR NO SEPARATION IN WATER DRUM


CIRCULATION IN FIRE ROW TUBES FOR THE TUBE BANK
CONSIDERED FOR THE MAIN FRESNEL A-TYPE BOILER
N.B. CURVES ARE FOR 12 LBS OF OIL/SQ FT OF B.H.S. FOR OTHER BOILERS
DEPTH OF TUBE BANK REDUCED BY OMISSION OF TUBES NEAR TO UPTAKES.
NO EXTERNAL DOWNCOMERS USED
NOMENCLATURE:-
ρ_t = DENSITY STEAM/WATER MIXES
ρ_w = DENSITY OF WATER @ SAT TEMP
$\frac{\rho_t}{\rho_w}$ (DOWNCOMERS) = 1.60 NO SEPARATION IN WATER DRUMS
$\frac{\rho_t}{\rho_w}$ (DOWNCOMERS) = 1.80 NO SEPARATION IN WATER DRUMS
$\frac{\rho_t}{\rho_w}$ (DOWNCOMERS) = 1.00 WITH SEPARATION IN WATER DRUMS
$\frac{\rho_t}{\rho_w}$ (DOWNCOMERS) = 1.80 WITH SEPARATION IN WATER DRUMS
$\frac{\rho_t}{\rho_w}$ (DOWNCOMERS) = 1.80 NO SEPARATION IN WATER DRUMS
$\frac{\rho_t}{\rho_w}$ (DOWNCOMERS) = 1.00 WITH SEPARATION IN WATER DRUMS
$\frac{\rho_t}{\rho_w}$ (DOWNCOMERS) = 1.80 WITH SEPARATION IN WATER DRUMS
MEAN WEIGHT OF MIXTURE OF STEAM & WATER PER CU FT
MEAN DENSITY OF MIXTURE
NO OF TUBE ROWS IN TUBE BANK
APPENDIX D Fig 6


CIRCULATION IN TUBE ROW NEAREST THE FIRE FOR THE TUBE BANK CONSIDERED FOR THE WATER TUBE A-TYPE BOILER
No. of Tube Rows in Tube Bank


Circulation in Fire Row Tubes of The White-Forster A-Type Boiler.
(AREA EXTERNAL DOWNCOMERS)/(AREA THRU FIRE RAWS)

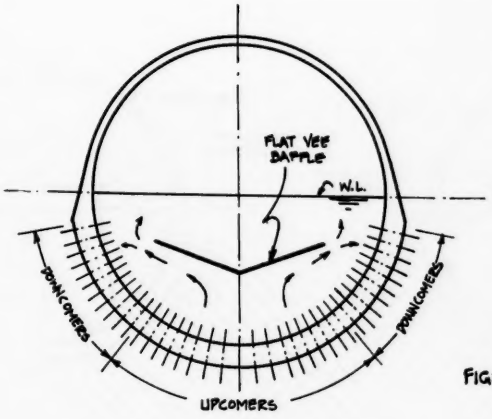


FIGURE 1

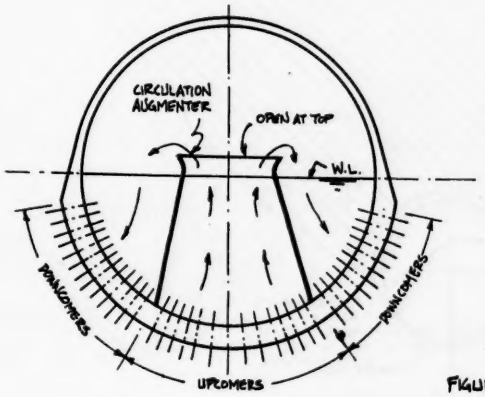
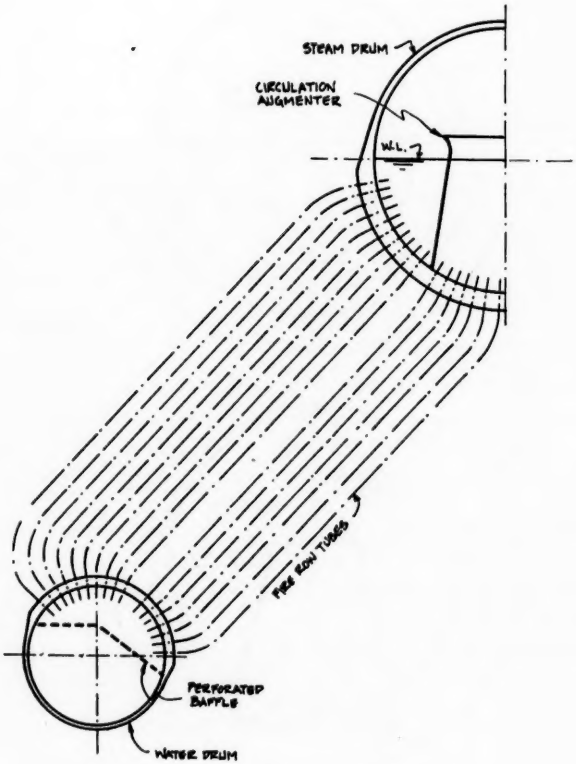
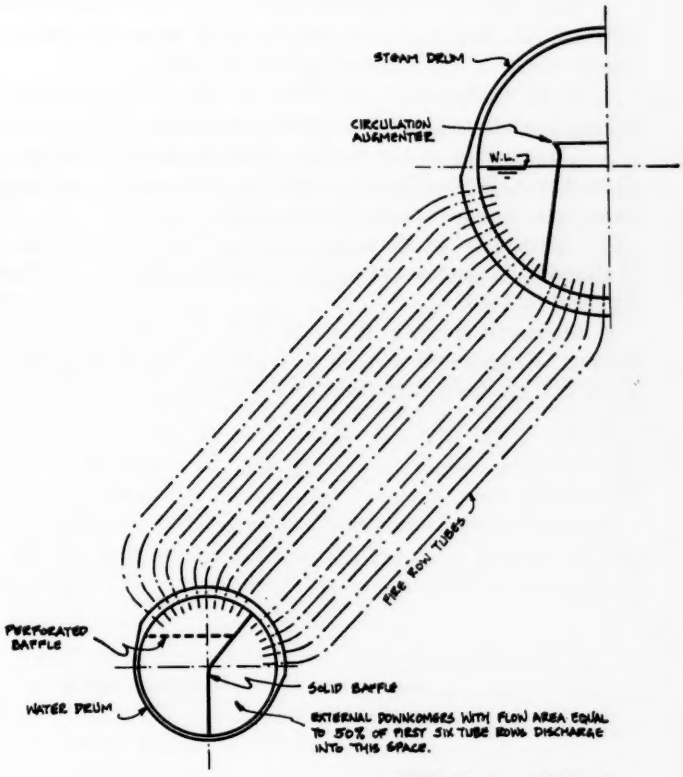


FIGURE 2

APPENDIX E — FIGURES 1 & 2



APPENDIX E – FIGURE 3



APPENDIX E – FIGURE 4

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3. R. M. Hanson on "Circulation in Water Tube Boilers" in Edison Electric Institute Bulletin for September, 1936.

4. "Description of Experiments Made at Messrs. Yarrow and Co.'s Works, Poplar, January, 1896" in Journal of the American Society of Naval Engineers, 1896.

5. "Circulation in Water-tube Boilers," by Professor W. H. Watkinson, in Journal of the American Society of Naval Engineers, 1896.

6. Engineer-Captain S. R. Dight, R.N., on "Naval Water-tube Boilers. Experiments and Shop Trials," in the Transactions of Institution of Naval Architects, 1933.

THE MATERIALS CORROSION INVESTIGATION AT EASTPORT, ME.

By W. J. Jeffries, Member.*

Among the several comprehensive investigations of corrosion in sea water now underway, the Materials Corrosion Investigation at "Quoddy" is important, because of the great variety of materials under test. Moreover, the temperature of the water is low and the fouling season short. The brief summary here presented is valuable. The quantitative measurements will give additional data of value when the tests are completed.

For the purpose of determining the most suitable metallic and non-metallic materials for use under severe sea water service in the construction of the Passamaquoddy dam, the U. S. Engineers Office at Eastport, Me., inaugurated a program designated the "Materials Corrosion Investigation." Since its initiation in February, 1936, nearly 700 specimens have been exposed in the test.

All tests are made in triplicate; that is, in air, at mean tide, and submerged. The metals under test are classified as follows:

Class

1. Brass or Bronze.
2. Aluminum Bronze.
3. Bronze (Sn Bronze with Pb or Ni).
4. Silicon Bronze (Cu-Si Alloy).
5. Manganese Bronze (Cu-Zn with Mn).
6. Bronze (Zn bronze with or without Pb).
7. Nickel Copper.
8. Pure Metals (cu and Zn).
9. Non-Metallic Materials.

* Senior Materials Engineer, Standards Branch, Bureau of Engineering and Bureau of Construction and Repair, Navy Department, Washington, D. C.

10. Cast Iron.
11. Alloy Iron.
12. Steel.
13. Cast Steel.
14. Welded Materials (Steel).
15. Alloy Steel.
16. Corrosion-Resisting Steel (Cr types).
17. Corrosion-Resisting Steel (Cr-Ni type).

The exposure tests are being made at the Government wharf, Treat Island, Eastport, Me. The submerged specimens are exposed 25 feet below mean low water; 43 feet below mean high water. The specimens exposed in air are mounted 12 feet above mean high water and were only rarely, and then but slightly, wet by spray. The half tide specimens were almost always wet and a few of them are sometimes partially covered by vegetation. There is no pollution in the locality effecting the test.

The standard specimens made from cylindrical bars 4 inches long and 1½ inches diameter were cut longitudinally to give one rough machined surface and one smooth machined surface. The remainder of the surface contained the original mill scale. The surface area exposed was approximately 23 square inches. Three holes are provided for supporting and coupling the specimens together. Each specimen is mounted on a bakelite rod with bakelite spacers to hold the specimen midway in a wooden frame. Coupled specimens exposed for electrolytic corrosion consist of two standard specimens of dissimilar metals clamped together by steel bolts insulated from specimens by bakelite bushings. Specimens and couples are mounted on wooden frames on 12-inch centers providing a minimum distance of 8 inches between specimens in order to avoid corrosion influence between dissimilar metals in adjacent positions.

Two types of inspection are made: (*a*) visual inspection, (*b*) quantitative inspection (photographed, cleaned, weighed and again photographed before re-exposure). Visual inspections were made during the first period of exposure (1936-1937) and semi-annually during the second period (1937-1938). The first quantitative inspection was made in 1937 at the end of the first period of expo-

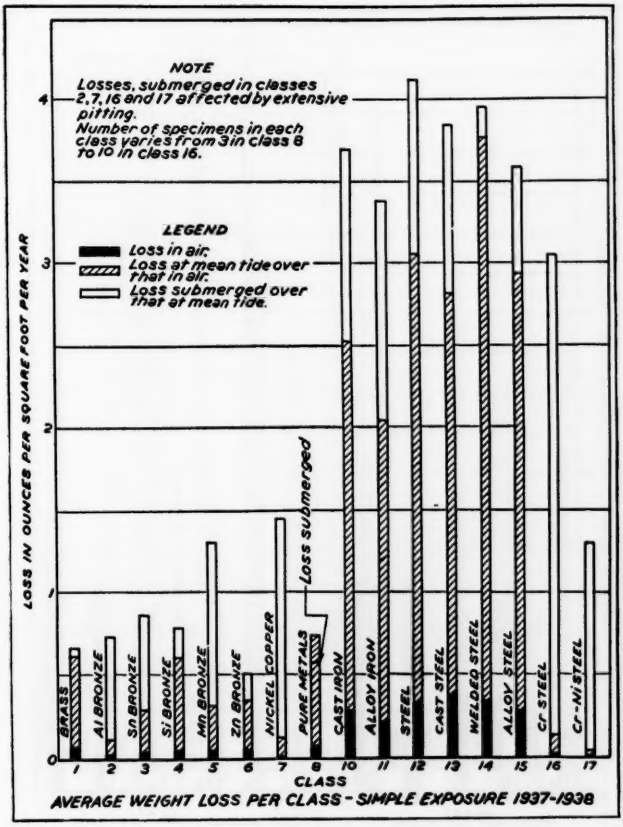
NOTE
Losses, submerged in classes 2, 7, 16 and 17 affected by extensive pitting.
Number of specimens in each class varies from 3 in class 8 to 10 in class 16.
LEGEND
Loss in air.
Loss at mean tide over that in air.
Loss submerged over that at mean tide.
LOSS IN OUNCES PER SQUARE FOOT PER YEAR
0
1
2
3
4
Loss submerged
BRASS
Al BRONZE
Sn BRONZE
Si BRONZE
Mn BRONZE
Zn BRONZE
NICKEL COPPER
PURE METALS
CAST IRON
ALLOY IRON
STEEL
CAST STEEL
WELDED STEEL
ALLOY STEEL
Cr STEEL
Cr-Ni STEEL
1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17
CLASS
AVERAGE WEIGHT LOSS PER CLASS - SIMPLE EXPOSURE 1937-1938

Figure 1

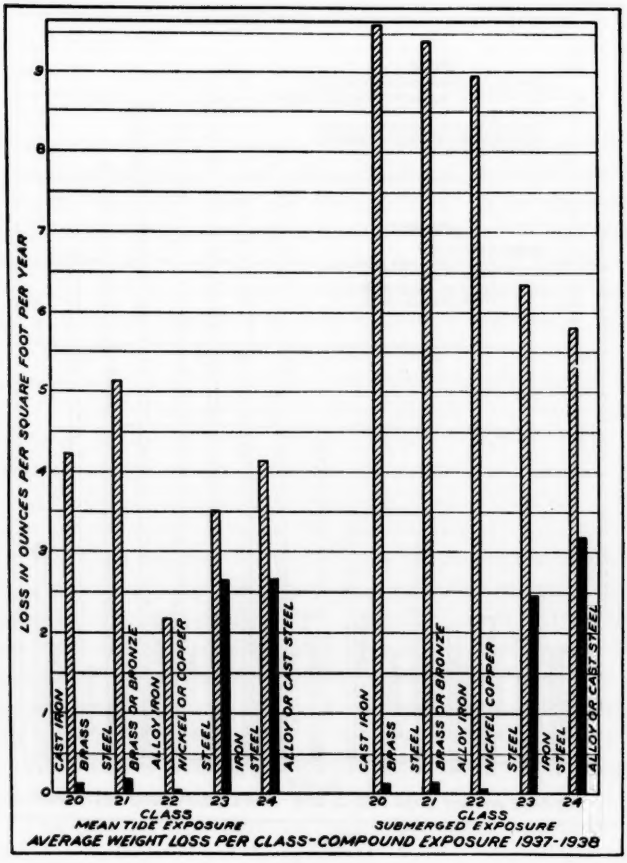
LOSS IN OUNCES PER SQUARE FOOT PER YEAR
0
1
2
3
4
5
6
7
8
9
CAST IRON
BRASS
STEEL
BRASS OR BRONZE
ALLOY IRON
NICKEL OR COPPER
STEEL
IRON
STEEL
ALLOY OR CAST STEEL
20
21
22
23
24
CLASS
MEAN TIDE EXPOSURE
CAST IRON
BRASS
STEEL
BRASS OR BRONZE
ALLOY IRON
NICKEL COPPER
STEEL
IRON
STEEL
ALLOY OR CAST STEEL
20
21
22
23
24
CLASS
SUBMERGED EXPOSURE
AVERAGE WEIGHT LOSS PER CLASS-COMPOUND EXPOSURE 1937-1938

Figure 2

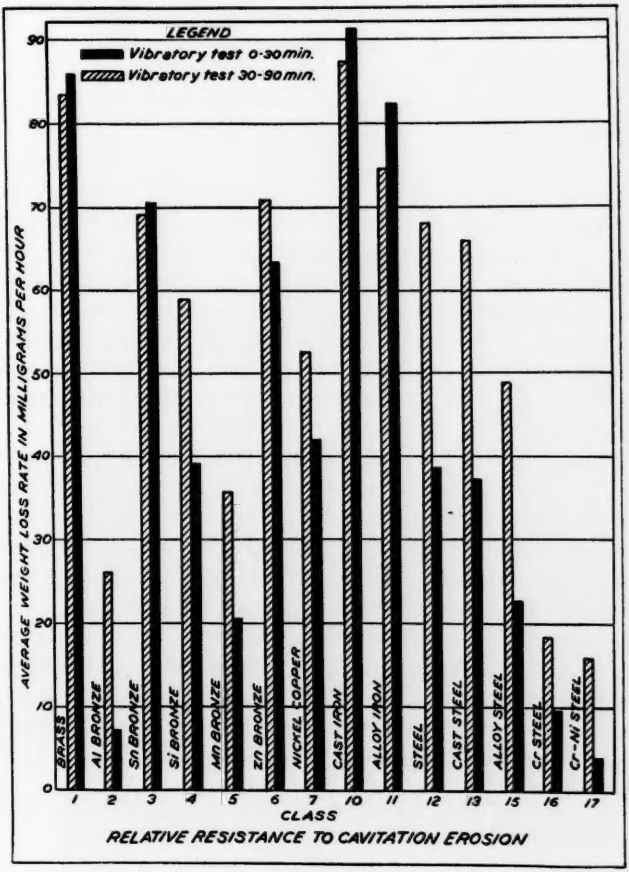
LEGEND
Vibratory test 0-30 min.
Vibratory test 30-90 min.
AVERAGE WEIGHT LOSS RATE IN MILLIGRAMS PER HOUR
0
10
20
30
40
50
60
70
80
90
BRASS
Al BRONZE
Sn BRONZE
Si BRONZE
Mn BRONZE
Zn BRONZE
NICKEL COPPER
CAST IRON
ALLOY IRON
STEEL
CAST STEEL
ALLOY STEEL
Cr STEEL
Cr-Ni STEEL
1
2
3
4
5
6
7
10
11
12
13
15
16
17
CLASS
RELATIVE RESISTANCE TO CAVITATION EROSION

Figure 3

sure. The second quantitative inspection was made in 1938 at the end of the second period of exposure. In the future it is planned to make visual inspections annually, with quantative inspections at the end of the fifth and tenth years. Notwithstanding the discontinuance of the U. S. Engineer Office at Eastport, Me., on November 1, 1936, the Chief of Engineers directed the continuance of the investigation under the U. S. Engineer Office at Boston, Mass.

As the exposure tests have not been completed, it is felt that it would be unwise to draw any conclusions at this time. However, the results to date are of interest and value if properly interpreted. They indicate trends which may be considered representative, particularly as to results by classes, rather than for individual specimens. It is manifestly absurd to expect conclusive results as to relative corrosion resistance of metals differing but little in composition when only one sample of each particular composition has been tested. Obviously, a large number of check tests would be required for that purpose. Caution is particularly advised in the application of particular results in detail to other environmental conditions.

Figure 1 shows the relative order of resistance to corrosion in simple exposure 1937-38 in air, at mean tide, and submerged. Figure 2 shows the effect of electrolytic action on the rates of corrosion on dissimilar metals in contact in compound exposure 1937-1938 at mean tide and submerged. The relative rates measured 1936-1937 are similar to those shown for 1937-1938.

A serious complication in selection of metals for sea water service is involved if possibility of cavitation must be considered. Figure 3 shows the relative order of resistance to cavitation erosion in sea water as determined from vibratory tests made at the Massachusetts Institute of Technology (Trans. A. S. M. E., July, 1937).

NOTES.

INDEX TO NOTES.

Flange Bolts for High-Pressure, High-Temperature Steam Piping.
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FLANGE BOLTS FOR HIGH-PRESSURE, HIGH-TEMPERATURE STEAM PIPING.

This article is concerned with the selection of materials and the forms of bolts and flanges for the subject application. The author is R. B. Büchele. The original was published in German in the July 22, 1939, issue of Die Warme, Berlin, Germany. The translation and rewriting is by R. Michel, Marine Engineer, New Design Branch, Design Division, Bureau of Engineering and Bureau of Construction and Repair, Navy Department, Washington, D. C.

I. INTRODUCTION.

The successful use of various types of flanged joints in high-pressure, high-temperature steam piping is dependent upon the correct use of bolts. These should be capable of withstanding tensile stresses of about 18,500 pounds per square inch for long periods of time and additionally be able to withstand impact loading of such magnitude that the total load approaches the allowable creep limit. This loading should leave the bolts still capable of being operated within the elastic limit.

Furthermore, a flanged joint should be so constructed that it may be readily unbolted. It is therefore necessary that the bolt threads release the nuts after long periods of operation and also remain serviceable.

II. CONCERNING MATERIALS.

Various materials which meet the above requirements for flange bolts are obtainable in the markets. Suitable alloys for this purpose are chromium, molybdenum, vanadium and silicon.

Nickel has been omitted as an alloy material because it causes unfavorable changes in the steel structure when a combination of high temperature and tensile stresses exists for long periods of time.

Suitable steels possess the following properties:

Tensile strength, pounds per square inch	113,000-135,000 at 70 degrees F. 100,000-107,000 at 572 degrees F.
Yield point, pounds per square inch	85,300 at 70 degrees F. 74,000-77,000 at 572 degrees F.
Creep limit, pounds per square inch	64,000 at 752 degrees F. 28,400-42,700 at 932 degrees F.

Elongation in 5 diameters length at 70 degrees F. = 16 per cent.
Reduction in area at 70 degrees F. = 50-60 per cent.

Figure 1 shows curves giving the above properties and also includes modulus of elasticity and thermal expansion. The curves lettered "c" and "d" give the minimum and maximum loads for which these bolts are suitable.

Resistance to impact is of special importance in the case of a bolt. This is determined by an impact test. Steels of the above composition have impact resistances of 373 to 560 foot pounds per square inch. In case a test specimen has been subjected to a long-time tensile test at high temperature and is then given an impact test, failure may show an irregular, inter-crystalline fracture. Room-temperature tensile tests of a specimen which has been previously subjected to a long-time elevated-temperature test may, however, show high values of tensile strength, yield point, elongation and reduction of area. Only an impact test will show whether the material is suitable for high-temperature, high-pressure piping service.

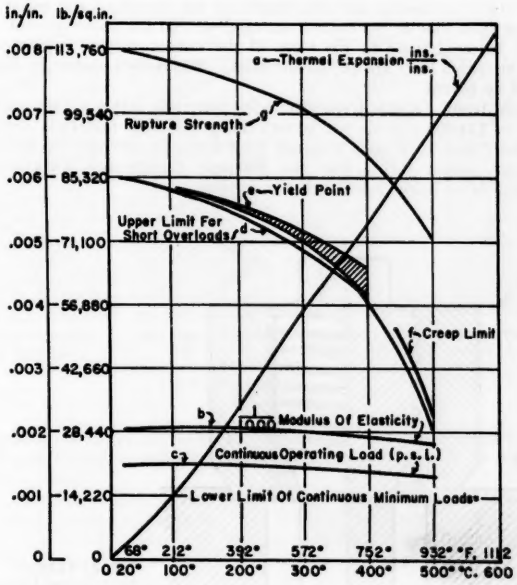


FIG. 1. PROPERTIES OF BOLT MATERIALS.

An explanation of the behavior of certain alloy steels formerly used for high-temperature, high-pressure service will be given below. Suffice it to say at present that the inter-crystalline material is changed by segregation and that the addition of nickel is favorable to this change.

A further requirement of a satisfactory bolt material is that it must have good resistance to repeated stresses.

In the past, desirable properties were determined only by means of long-time tensile tests. These were generally conducted on a notched tension specimen at 932 degrees F. Fractures did not occur after 5,000 hours of operation when the smallest cross section at the notch was subjected to a load of 35,600 to 41,200 pounds per square inch. For the time being we consider only as suitable for the subject service such bolt materials which have passed this long-time tensile test.

III. MATERIALS SUITABLE FOR THREADING.

Easy unscrewing of the threads may be assured by: (1) sufficient clearance at the tips of the threads so that space remains for deformation, (2) a difference in the strength of the bolt and nut materials which assures that the softer thread takes the form of the harder thread material. High accuracy of manufacture of the threads is necessary to conform with these requirements.

It is further necessary that the length of the thread of the softer material be shorter than that of the harder material so that the softer thread will have a chance to mold itself to the form of the harder thread, regardless of the position in which they are bolted together. The design shown in Figure 2 is protected by patent.

As seen from Figure 1, proper bolt materials have a minimum tensile strength of 114,000 pounds per square inch. Suitable materials for nuts following the design of Figure 2 should have a tensile strength of about 150,000 pounds per square inch. For this purpose chrome-molybdenum-vanadium steels, capable of being machined, are suitable.

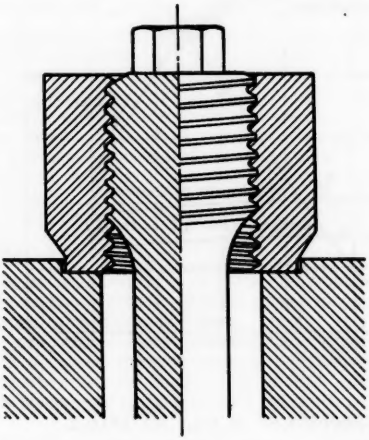


FIG. 2. FLANGE BOLT DRGM. USED, ℄ IN JOINT SHOWN IN FIG. 7.

The advantage of nuts, in accordance with Figure 2, over those of older designs which had a tensile strength of about 100,000 pounds per square inch, will be obvious. The turned portion at the base of the nut is to center the bolt in the hole.

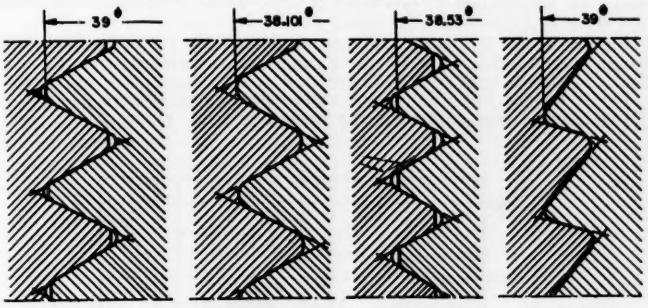
IV. THREADS.

The threads best adapted for the purpose are the Whitworth DIN 11, the Metric DIN 13 and 14, and the Whitworth fine-thread No. 2, DIN 240. Figure 3 shows these threads for a 1½-inch diameter bolt.

Experience has shown that the Whitworth fine-thread No. 2 cut in softer nuts of old design have a tendency to seize, probably because the first thread in the nut, which carries the largest load, has too small a load-carrying surface.

The Metric thread has a sharp cornered root which tends toward stress concentration in the bolt. In the case of the Whitworth bolt thread, DIN 11, the great depth results in a large diminution of the bolt root diameter, which must be increased in order to obviate the possibility of the elastic work being carried by the threaded portion of the bolt. The latter may cause high stress concentration at the threads.

FIG. 3. THREADS FOR 1 1/2" DIAMETER BOLT.



A common characteristic of these three thread forms is that they are symmetrical and, therefore, offer the same resistance to a tensile as well as to a compressive load.

This symmetrical form of thread is neither necessary nor desirable in the case of highly loaded screw bolts. The form of the thread should be designed to resist only tensile loads. For this purpose the stepped thread of Figure 3 is offered. Proportions of this thread are as follows:

Outside diameter $= D_o$.
Pitch $= S = 0.1D_o$.
Double depth of thread $= t = 0.5S$.
Radius $= r = 0.1S$.
Pitch diameter $= D_t = D_o - t$.
Root diameter $= D_k = D_o - S$.
Thread angle $= 14$ degrees 20 feet $+ 50$ degrees $= 64$ degrees 20 feet.

With reference to present practice, D_o should have the following values in millimeters: 24, 27, 30, 33, 36, 39, 42, 45, 48. The play at the thread tips is chosen to suit. Independent of this a certain clearance is allowed for thread deformations, lubrication and for the venting of air from cotter pin holes.

The stepped thread form as above described is protected by patent.

V. LOAD-CARRYING CAPACITY OF BOLTS.

The German DIN standards determine the bolt size with respect to allowable play between the outside thread diameter and the hole diameter. The bolt shaft diameter is also determined by these standards. It is clear that the shaft of the bolt is the working part and that the shaft diameter must therefore be smaller than the root diameter of the thread so that the elastic work is not carried over into the threaded portion, which would cause (1) unreasonable deformation of threads, (2) stress concentration with consequent danger of fracture under repeated loads at the thread roots.

We will now examine the load-carrying capacity of the bolt shaft. Figure 4 gives a variation in length of a bolt shaft 3.937 inches long, corresponding

to the loads obtaining at the various temperatures of Figure 1. Figure 4 also gives explanation of two types of alloy steels, viz.: molybdenum alloy and chrome-molybdenum alloy steels.

Figures 5 and 6 give the minimum load limit for a bolt having a shaft length of 11.8 inches. The following examples are included:

(1) A flanged joint, Van Stone narrow-flange type—Figure 7.
(2) Flanged joint with wide Van Stone pipe ends—Figure 8.

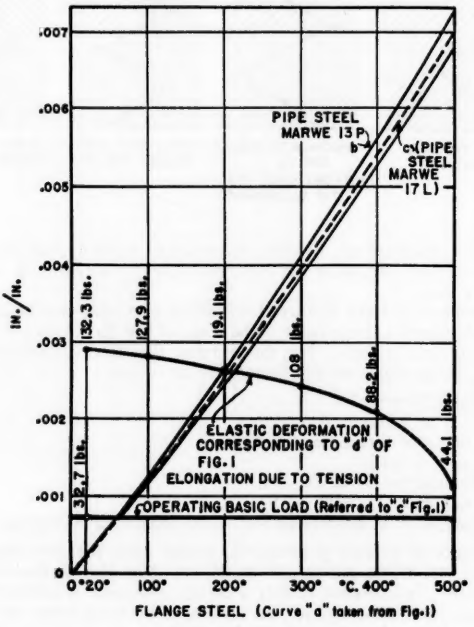


FIG. 4. THERMAL EXPANSION.

It is recognized that due to the unequal heating of the flanged joint the following occurs relative to the loading of the bolts:

(1) The bolt loading will be less for the type of connection shown in Figure 7 than that for Figure 8.
(2) The use of molybdenum or copper alloyed steels with the construction shown in Figure 8 is particularly disadvantageous whereas such steels may be used with the design shown in Figure 7.
(3) The load-carrying capacity of the bolts is dependent upon the length of the bolt with reference to diameter of upset pipe end.

(4) In order to insure greatest carrying capacity of the bolts it is recommended that the following fraction have a high value:

$$\frac{L}{2(B+F)}$$

Where L = shaft length of bolt, B = height of upset pipe end above outside pipe diameter, F = thickness of flange.

(5) Slow warming up of piping and adequate insulation are conducive to long life of the joint.

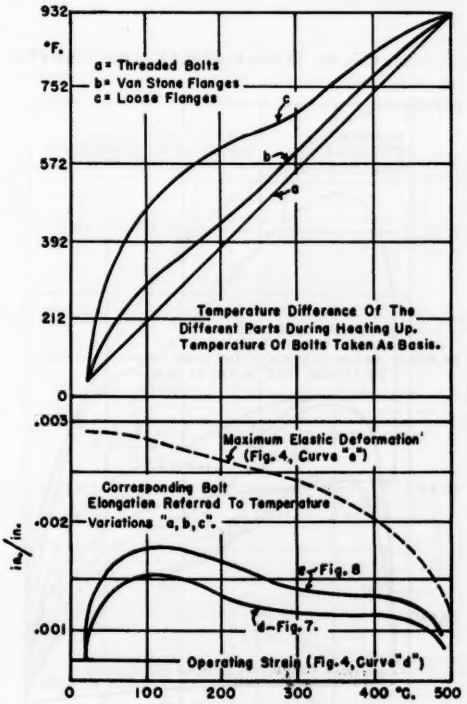


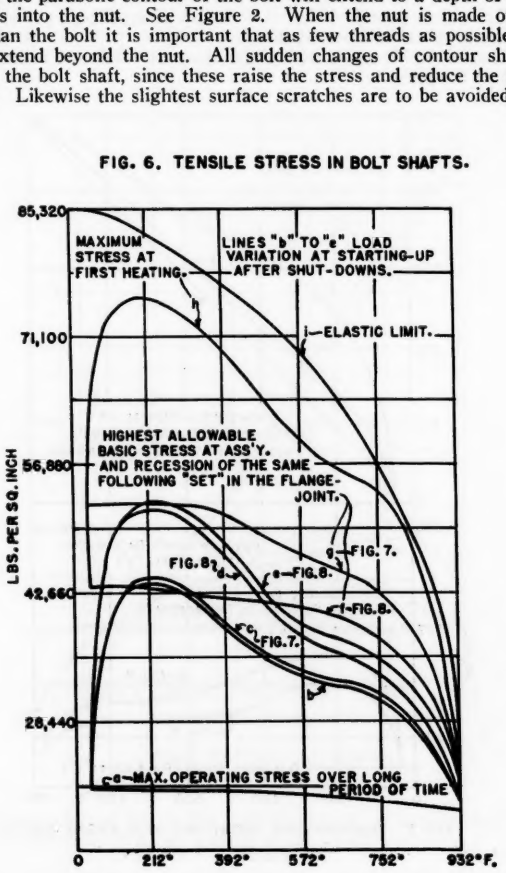
FIG. 5. TEMPERATURE VARIATIONS OF FLANGED JOINTS OF FIGURES 7 OR 8.

VI. DESIGN OF PARTS.

The usual practice is to make the bolt shaft of cylindrical form. Perhaps a more scientific design would be to make this slightly spindle or conical formed. However, the deviation from a cylindrical form would be so slight that any other form is impractical. Care must be taken that the bolt shaft is free of any notch defects due to any cause whatsoever.

For the transition from shaft to threaded portion of bolt a parabolic profile has long been recognized as the best. However, it is difficult to make a parabolic surface merge with that of the threads. It is therefore advisable to make the bolt diameter so much smaller than the root diameter of the thread that the parabolic contour of the bolt will extend to a depth of at least two threads into the nut. See Figure 2. When the nut is made of softer material than the bolt it is important that as few threads as possible of the bolt will extend beyond the nut. All sudden changes of contour should be avoided on the bolt shaft, since these raise the stress and reduce the possible elongation. Likewise the slightest surface scratches are to be avoided on the bolt shaft.

FIG. 6. TENSILE STRESS IN BOLT SHAFTS.



VII. INFLUENCE OF FLANGE DESIGN ON THE BOLTS.

Figures 5 and 6 show that the bolts are highly stressed.

It goes without saying that the loads on the flanges are correspondingly high. For the following reasons the flanges are subjected to bending stresses, the magnitudes of which have to date not been accurately calculated:

For a piping layout with essentially straight contours, which is the design striven for, the load-carrying capacity of the flanges, based on yield strength, should be greater than that of the bolts. If the thickness of the flanges is determined with this in mind, deformation of these is an elastic one, provided the bolts have not been stressed beyond the yield point. Among other things, it follows, in the case of flanges which are just thick enough to be stressed within the elastic limit, that the stresses within the bolts will be relieved. The determination of this reduction in load on the bolts should be accomplished. Up to date it has not been customary to do this because,

(1) In the case of joints at turbines or boilers the flanges are often made too thick.

(2) Elastic deformation of even thinner flanges is kept down to very small limits because of resulting high stresses at the inner flange diameter.

(3) All bending of flanges results in relieving the loads on the bolts on one side only.

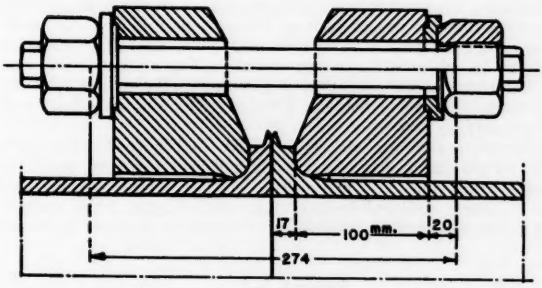


FIG. 7. FLANGE JOINT WITH FLANGES DRGM.

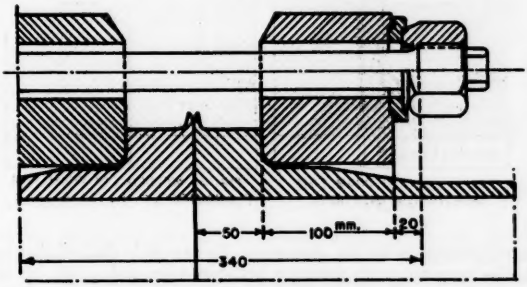


FIG. 8. FLANGE JOINT WITH UNIFORM THICKNESS FLANGES.

If the advantage of spring action of the flange is not considered by the designer in making calculations, he simplifies his calculations by selecting flange thicknesses which will assure ample rigidity.

It will, nevertheless, always be of advantage if at least one side of the flanged joint is provided with a gasket which can be rotated and whose

elastic deformations avoid the occurrence of high localized stresses such as might exist at the throat of flanges. Preferred material for loose flanges is oil-quenched chrome-molybdenum steel having room temperature tensile strength of 99,500 to 114,000 pounds per square inch and a high elongation, but a low creep limit (14,200 pounds per square inch at 932 degrees F.). Such a flange will assume its original position after bending out of shape by momentary large bolt loads; however, under continued bending stresses at high temperature, such a flange will deflect sufficiently so that the bolts will not be subjected to a load exceeding the endurance limit.

The relief of the initial stress due to pulling up of the bolts is assured by this deflection of the flanges. Figures 5 and 6 show that the provision of additional length of the bolts is an advantage in case the bolt thread is shorter than the thread of the nut. The additional length of bolts required for a joint such as is shown in Figure 8 is not of advantage from the viewpoint of endurance strength of the bolts. Upon heating up, the upset ends of the pipe are first elevated in temperature. The greater the width of the flanged pipe ends, therefore, the greater will be the expansion of the bolts. It is therefore advantageous that the design such as Figure 7 be followed. It would be too much to expect the gasket to be deformed under operating conditions. The gasket must be subjected to enough initial loading so that it will not become further deformed under operating conditions. This means that the gasket must be made of material that has a sufficiently high creep limit and also be corrosion resisting.

A preferable design for gaskets is shown in Figure 9. The surfaces are serrated and the material is of corrosion and heat resisting steel. The outer periphery of this ring is so made that the ring is automatically centered at erection. Gaskets designed in accordance with Figure 9 are suitable for all kinds of piping material and loadings. Graphite is inserted in the serrations

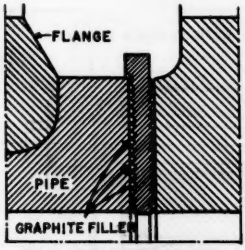


FIG. 9. PACKING DRGM. FOR FLANGE JOINT, e.g., FIG. 7.

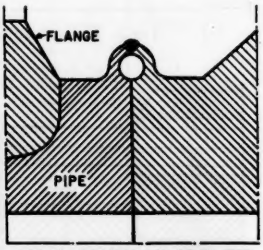


FIG. 10. WELDED LIP DRGM. FOR FLANGE JOINT, e.g., FIG. 7.

as corrosion protection for the pipe ends. In case it is desired that no gasket be used, the required unit pressure between pipe ends may be calculated from the cross-sectional area of the bolts and the corresponding flange thicknesses. A third design is shown in Figure 10. The welded lips are capable of making a tight joint even in case flat pipe surfaces do not seal off the steam.

VIII. PROTECTION AGAINST OVERLOAD OF BOLTS.

The correct design of the flanges will make it impossible to overload the bolts. If the flanges are elastically deflected, the bolt shafts will be unevenly stressed. Uneven stressing of the bolts will have a detrimental effect on their

endurance. Consideration must also be given to errors in workmanship. For example, the bearing surfaces of the bolt heads and nuts may not be at right angles to the axes of the bolts, etc. Making these surfaces spherical will not remedy the trouble because of the great amount of friction between the spherical surfaces.

Another solution is offered. If the washers underneath the nut and bolt heads are made of high carbon steel they will have enough strength in their cold condition to withstand the bolt pressure. Under operating temperatures the washers will become deformed. The result will be that the stress in the bolts will drop to the originally intended amount, and all of the bolt fibers will be equally stressed. Furthermore, all bolts of a particular joint will be equally stressed even though varying amounts were initially imposed on the bolts. Sometimes the bolts in a particular flange joint do not originate from the same melt of steel and therefore will be stressed varying amounts for equal pulling up of the wrench. In all of these cases when the surfaces between nuts and washers are subjected to repeated loads the deformed surfaces will cause the loads on the bolts to be a minimum.

IX. SPRINGS.

The correct design of flanged joints is dependent upon the recognition that the bolts used therein are springs which were initially stressed and which adjust themselves to large operating loads by a change in length. It is possible, by the application of special elastic elements, to minimize the difficulty of correctly fitting the bolts to the flanges. This thought will be further developed in a subsequent paper.

It has been shown that the flanges act as plate springs when the material has been advantageously chosen. The problem of avoiding overloading of particular bolts in a flanged joint may be considered solved. Calculation of stresses within bolts is accomplished by the deformation of washers placed under bolt heads and nuts.

A flanged joint is subject to singular loadings, however, which result from the unequal heating up of the pipe line. In such case elastic deformation within the flanged joint is instrumental in causing leaks. Displacements of the pipe line must be taken up within the pipe itself. This can occur only if the flanged joint is stronger than the pipe was before being acted upon by the disturbing force. Under these circumstances the stress in the bolts must not exceed the yield point, even though the stress in the pipe has exceeded the yield point of the pipe material. In addition, the minimum pressure necessary to avoid leakage must be maintained. Nevertheless, it is possible to imagine pipe layouts in which the elastic action of the flanges and bolts is of advantage.

X. BOLT ASSEMBLY.

In case the bolts and nuts are carelessly assembled, undue stress of the bolt shafts may result. This excessive stressing may be particularly undesirable with the aging of certain bolt alloy materials. It is important that the steels used have properties such that elongation of the bolt while cold will not be detrimental to the elastic properties of the bolt after long service.

It is of course desirable that the elongation of the bolts be measured at assembly. However, this is not as easy as might appear.

It must first be kept in mind that an accurate measurement insrument must be used to make measurements on a fairly rough machine job. If, for example, the pitch of the thread is 0.1417 inch, it does not take a large

rotation of the nut to produce an elongation of 0.00039 inch. The large forces which are called into play in tightening up the nuts, however, cause only a backward motion on the nut and the bolt. This causes:

(1) An elastic stretching of the bolt.
(2) An elastic bending of the flanges.
(3) Deformation of the threads.
(4) Deformation of the contact surface of the nut.
(5) An adjustment of all contact surfaces, etc.

All of the above circumstances are varied somewhat with different bolts and in any one bolt the fitting of the various contact surfaces is of great importance.

It will therefore be impossible in the case of a joint made up of 12 to 16 bolts to attain a uniform, initial stress in each by merely measuring the elongation. It is probable that conditions in the strength of the materials making up the various bolts will be great enough to cause different stresses in the bolts even though the same elongation is measured on each one.

Better success in tightening up bolts has been attained by experienced workmen by sensing the pull on the wrench, rather than by the use of measuring instruments.

The following should be added: All contact surfaces in a flanged joint are more or less rough. The high spots of the rough surfaces contact each other. After being in service for some time these high spots are reduced, this generally occurring during the first long period of full-temperature operation of the joint. The result of this is that a readjustment or setting of the various parts takes place. This reduces the stress in the bolts to a point very much under the initial stress given them. For this reason flange joints will often become leaky a short time after operation, even though they were initially drawn up to a theoretically correct stress in the bolts. It is therefore necessary to provide a considerable excess initial stress in tightening up the bolts in order that the joints may remain tight after many thousands of hours of operation. Experience has shown that this additional load on the bolts does no harm to the bolt material when all other conditions of correct design have been specified. The initial set that occurs in all of the surfaces under pressure, makes it desirable that an initial overloading of the bolts be effected and in addition it automatically reduces this initial overloading.

XI. THE EXPECTED LIFE OF FLANGE BOLTS BASED UPON PAST EXPERIENCE.

In answer to the question: How long may a flange bolt be considered serviceable? the following table is offered:

CHROME-NICKEL-MOLYBDENUM STEEL.
(Considered unsuitable for high-temperature bolt material.)

Creep Limit at 930 Degrees F.	*Long-Time Tensile Test on Notched Rods*		*Service Hours of Threaded Bolts, Threads Under Tensile Loading*
	Stress at Notch	*Service Period*	
40,000 pounds per square inch.	28,400 pounds per square inch.	700 hours.	Greater than 27,000 hours, temperature 914-950 degrees F.
	22,750 pounds per square inch.	1400 hours.	Load at root of thread = 11,000 to 12,800 pounds per square inch.

CHROME-MOLYBDENUM STEEL.
(Considered suitable for high-temperature service.)

Creep Limit at 930 Degrees F.	*Long-Time Tensile Test on Notched Rods*		*Service Hours of Threaded Bolts, Threads Under Tensile Loading*
	Stress at Notch	*Service Period*	
40,000 pounds per square inch.	40,000 pounds per square inch.	Greater than 5000 hours.	Greater than 80,000 hours to be expected from properly designed joint.

Bolts designed in accordance with Figure 2 would have a life equal to that of other important parts of a pipe line.

XII. SUMMARY.

According to the viewpoint of the author, the best design of high-pressure, high-temperature flanges and bolts is shown in Figure 7, and the best design of gasket in Figure 9 or 10.

Flanged joints made in accordance with Figures 7, 9 and 10 have proven themselves in service, some 2000 of these being used. Joints that were made steamtight at assembly have remained so for many thousand hours without any additional pulling up of the bolts. This has been the case when using 850 pounds to 1850 pounds at 930 degrees F. temperature, the operating experience extending to 40,000-50,000 hours.

Installations operating at lower pressures and temperatures and using flanges and bolts in accordance with the above designs by the firm Gesellschaft für Hochdruck-Rohrleitungen have been in service 10 to 12 years without leaking.

An answer to the question: How long may a flanged design in accordance with Figures 7, 9 and 10 be considered serviceable? cannot be given at present, since the service period has not been of sufficient duration to date.

SPECTROCHEMICAL ANALYSIS.

Quantitative chemical analysis by spectrograph is further developed by this article by Dr. A. E. Ruehle of the Bell Telephone Laboratories. The extract is from the January, 1940, issue of the Scientific Monthly, published for the American Association for the Advancement of Science, Smithsonian Institution Building, Washington, D. C.

When a substance is heated to incandescence or excited electrically, each of the elements present emits light of wave-lengths which are characteristic of that element. By analyzing the light emitted with a spectrograph, the nature and amount of the elements present can be found. Several thousand chemical determinations are made each year at the Laboratories by this method. It is an extremely sensitive method and will detect minute quantities of matter.

A sample of the material to be analyzed is heated to incandescence in the cup-shaped electrode of an electric arc whose light then takes on the character of the elements in the unknown substance. This light is split by a quartz prism into a spectrum and recorded as lines on a photographic plate. The

elements present in the sample are identified by comparing the positions of these lines with those of the known lines of the different elements. With a spectrograph of good dispersion, any metallic or metalloid element can be detected in a mixture.* Non-metals require special methods of excitation and the intensity of their spectra varies greatly with the nature of the mixture in which the element is found.

The intensity of a spectral line depends on many factors besides the abundance of the element in the source, but the experienced spectroscopist can make a fairly good estimate of the amount of each element present. This serves as a convenient guide for a later quantitative determination either by spectrochemical or ordinary methods.

The identification of contact materials is an excellent example of the technique of estimation. It permits finding of what materials a contact is made without removing the instrument from its mounting, without impairing the contact for further use, and without more than momentarily interrupting its service. The surface of the contact is rubbed with a small piece of pure abrasive paper, the paper is burned in a carbon arc and the resulting spectrum photographed. From the lines in this spectrum, which do not appear in that of the abrasive paper alone, the elements which compose the contact can be identified and their proportions estimated (Figure 3).

The most direct method of quantitative analysis is to compare the spectrum of the sample with the spectra of prepared standards taken under the same conditions and on the same photographic plate (Figure 4). It is generally difficult to prepare a series of homogeneous solid standards for this purpose. A more practical method is to put the sample in solution and to compare it with standard solutions of known concentration. Measured portions of each solution, dried on graphite electrodes, then serve as the test pieces for the analysis. This general method can be applied directly to a large number of cases without appreciable modification. Difficulties arise only when one or more components of the sample in question are not readily held in a reasonably concentrated solution. By this method the amount of an element present in the sample can be determined to within about 10 per cent.

Much progress has been made in devising methods of avoiding the use of standard spectra on every plate for routine work. For example, a curve which shows the difference in density of an impurity line and that of a line of some invariant component of a mixture, as the amount of impurity changes, can be used as a reference standard to determine the amount of the impurity in an unknown mixture. Changes within reasonable limits in excitation conditions or photographic technique will change the density of both the impurity line and the reference line in the same direction and to the same degree. This is called the internal standard method. It gives the most precise results yet attained in spectrochemical analysis, but to reach this precision it is necessary to calibrate the response of each photographic plate for different intensities of the spectral lines. This is conveniently done by a step-sector as shown in Figure 5. By rotating the sector in front of the slit, spectra are obtained with steps of different density. Plotting from densitometer readings the density of each step of a line against the logarithm of the exposure, *i.e.*, the logarithm of the angular apertures of the sector, gives a curve which shows how density of a line changes with intensity for that plate. Since the shape of the curve varies from plate to plate, the difference in density of two lines will not be comparable unless corrected for this variation. By taking these factors into account the precision of quantitative determinations has been more than doubled.

* Bell Laboratories Record, May, 1928, p. 289.

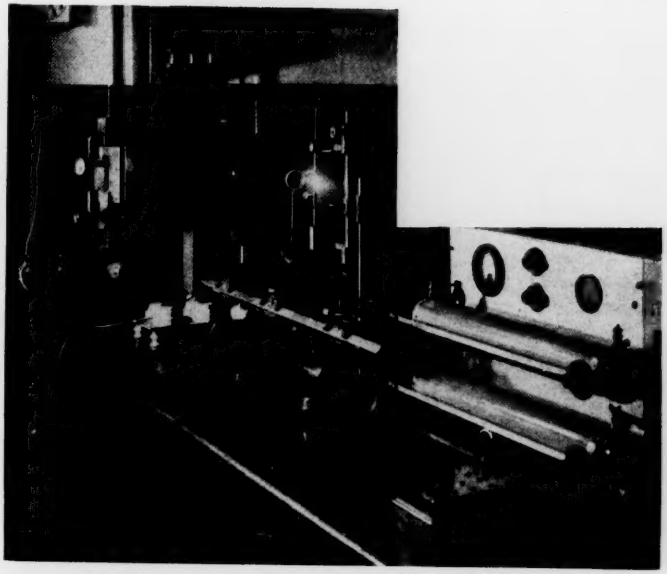


FIGURE 1. PHOTOGRAPHING AN ARC SPECTRUM.

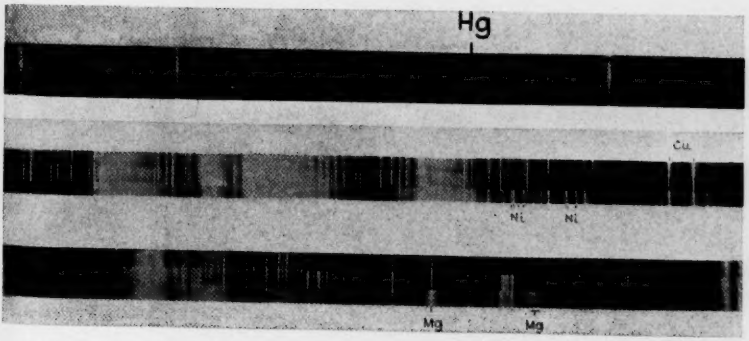


FIGURE 2. (ABOVE) DETECTION OF MERCURY AT THE POINT OF FAILURE OF ZINC SUPPORTS FOR AERIAL CABLES. (BELOW) DETECTION OF MINOR COMPONENTS, MAGNESIUM, NICKEL AND COPPER, IN ALUMINUM ALLOYS.

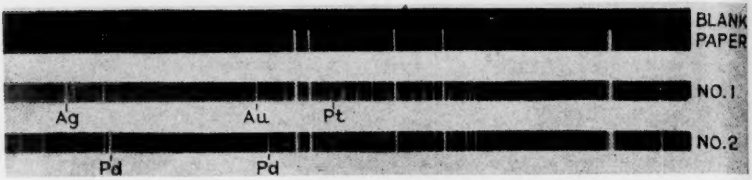


FIGURE 3. THE ELEMENTS IN CONTACT ALLOYS CAN BE IDENTIFIED WITHOUT DESTROYING THE CONTACT.

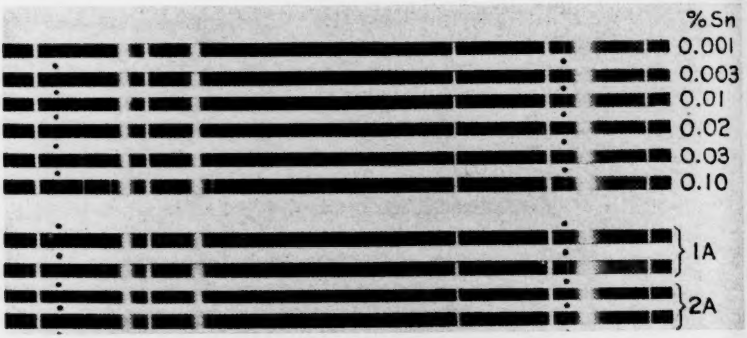


FIGURE 4. QUANTITATIVE DETERMINATION OF TIN IN CABLE SHEATH.

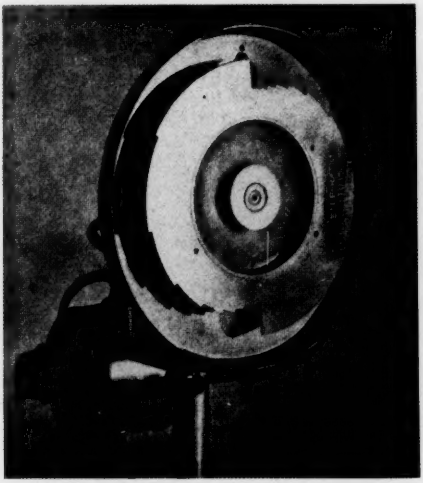


FIGURE 5. STEP-SECTOR WHICH IS ROTATED IN FRONT OF THE SPECTROGRAPH SLIT DURING AN EXPOSURE TO DETERMINE THE EXPOSURE-DENSITY CHARACTERISTIC OF THE PHOTOGRAPHIC PLATE FOR LIGHT OF DIFFERENT WAVE-LENGTHS.

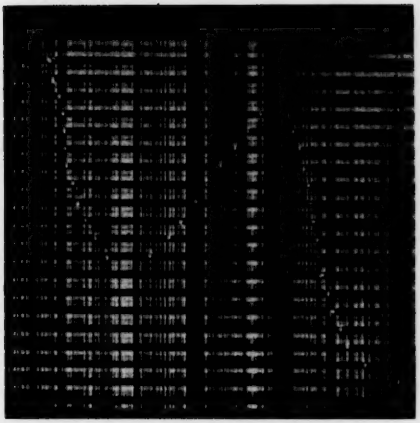


FIGURE 6. TYPICAL ABSORPTION SPECTRA. EXPOSURES ARE MADE IN PAIRS: ONE THROUGH THE ABSORPTION MEDIUM AND THE OTHER THROUGH A CALIBRATED DIAPHRAGM. BY MAKING A SERIES OF PAIRS WITH DIFFERENT DIAPHRAGM OPENINGS LINES CAN BE MATCHED IN DENSITY AS INDICATED BY THE DOTS.

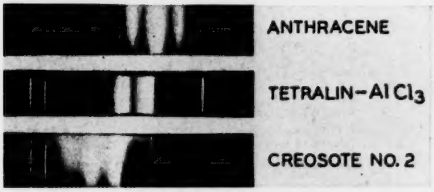


FIGURE 7. FLUORESCENCE SPECTRA EXCITED BY MERCURY LINES.

Many times the spectrochemical method can be applied as a probe when two presumably identical materials show unexpected differences in behavior. Nominally pure materials are never really pure and may introduce unsuspected impurities which vary with different lots of materials. The spectrograph shows differences in composition of the final product and thus often indicates the beneficial or detrimental effect of impurities.

Most spectroscopic analyses are made with emission spectra, but absorption spectra can also be used. If light of all wave-lengths passes through a solution, the amount absorbed will not be the same for all the different wavelengths; the spectrum of the emergent light will contain bands of different intensities. The amount of selective absorption may be used as a measure of the concentration of the dissolved substance (Figure 6). Advantage is taken of this phenomenon in the colorimeter and more specifically in the spectrophotometer, which employ visual matching to determine the amount of absorption. In a similar manner the spectrograph and a suitable photometer, used to measure the densities of the absorption spectra plates, find application for this type of analysis. Either visible or ultra-violet absorption bands can be used.

Another method of analyzing certain types of organic compounds is to measure the intensity of the fluorescent bands characteristic of the compound in question (Figure 7). In this case ultra-violet light is used to excite the fluorescent bands and the intensity of the bands is compared with that of the same bands in standard solutions of the same compound.

It is only in comparatively recent years that the spectrograph has come into widespread use for quantitative chemical analysis, but it has already proved itself a powerful tool, particularly for quick estimates of composition and where only minute quantities of material are available.

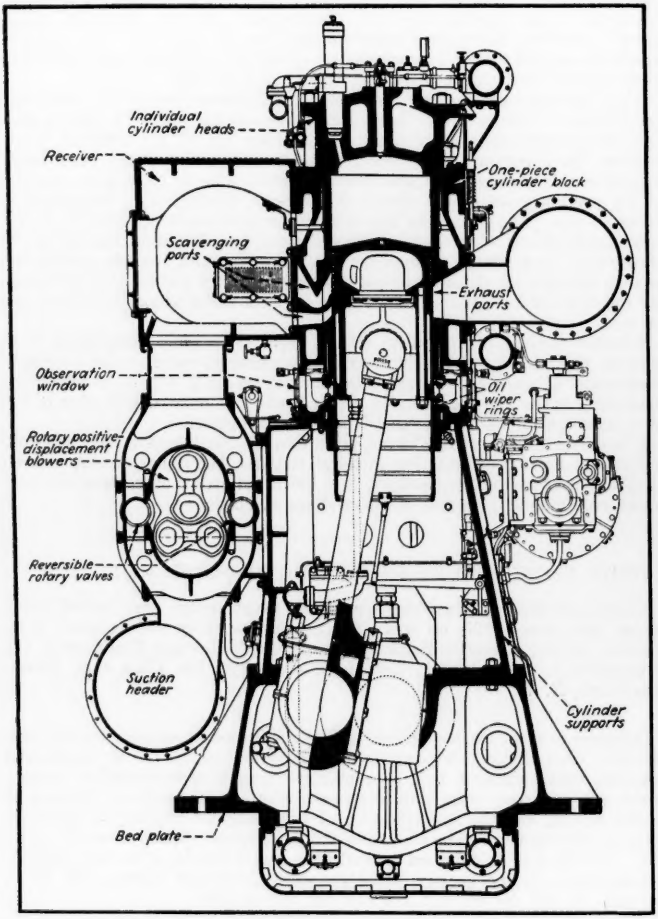
SHIP DIESELS BURN HEAVY OIL BY SOLID INJECTION.

Engines for the new Maritime Commission cargo vessels have burned heavy bunker fuels successfully on test. The article describing these engines is by Glenn C. Boyer, Associate Engineer, Burns and McDonnell Engineering Co. It appeared in the February, 1940, issue of Power, 330 West 42nd Street, New York, N. Y.

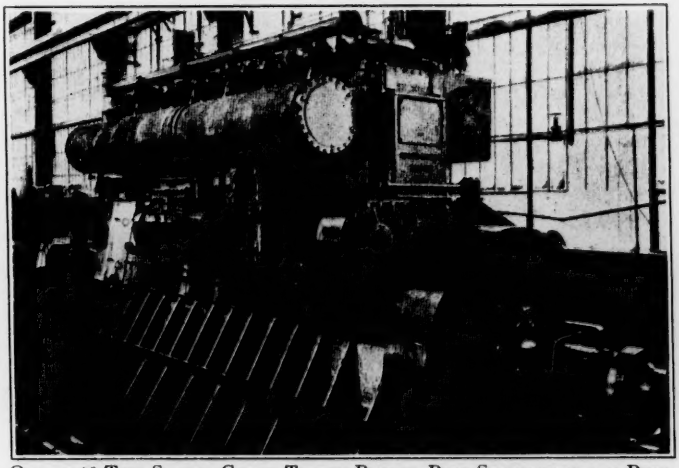
Designed to meet marine fuel-versatility requirements, engines for the new Maritime Commission C-3 cargo boats burn heavy residual oils by mechanical injection, thus opening a field of application long thought closed to engines of this type. Factory tests, plus experience with several stationary engines of similar design, indicate that efficiency of large engines measurably improves when use of solid injection eliminates the compressor required for air injection. On the test I witnessed recently the fuel analysis, after centrifuging, showed 0.993 specific gravity, 10.29 per cent Conradson carbon, 945 SSU. viscosity (at 100 degrees F.) and 3.02 per cent sulphur.

FOUR ENGINES DRIVE SHIP.

Sixteen engines of this type, being built by Busch-Sulzer Diesel Engine Co., will power four 8900-gross-ton cargo boats. Power plant for each ship consists of four Diesels driving a single screw through Westinghouse electromagnetic couplings and Falk reduction gears. At 85 shaft RPM. they will deliver a total of 8500 shaft HP. with continuous overload capacity of 10 per cent. Operation of all engines and couplings is centrally controlled.



SECTION THROUGH ENGINE (FROM FRONT) SHOWS FUEL INJECTION AND GOVERNING EQUIPMENT ON RIGHT SIDE; AND COMMON SUCTION AND SCAVENGING AIR HEADERS ON LEFT.



ONE OF 16 TWO-STROKE-CYCLE, TRUNK-PISTON, PORT-SCAVENGING AND PORT-EXHAUSTING, AIR-STARTING, DIRECT-REVERSIBLE, MECHANICAL-INJECTION ENGINES SET UP FOR TEST AT FACTORY.

Tests of 12 of the 16 engines indicate highly satisfactory performance, and while no official data on actual fuel rates is available, it is understood that they are approximately 0.38 pound per BHP. per hour at full load. Using the same set of injection nozzles for several engine tests gave a sufficient operating period to check difficulties that might be experienced with low-grade fuel. In addition, designers learned much from operation of a Busch-Sulzer mechanical-injection engine at Union Stockyards in Chicago, which has burned heavy oils for 7-8 years, and from another at the City of McMinville, Oregon (see table). Increased knowledge of injection principles and gradual improvement in design thus seem to have removed solid-injection fuel limitations.

COMPARISON OF FUELS BURNED IN TYPICAL HEAVY-OIL INSTALLATIONS

Fuel Sample	*1*	*2*	*3*	*4*
Heating value, Btu. per lb	18,845	18,775	18,315	18,690
Specific gravity, at 60 F	0.959	0.942	0.993	1.015
Flash point, F	211	206	261	201
Burning point, F	234	252	296	234
Acidity, %	0.018	0.030	0.004	0.037
Sulphur, %	0.94	1.24	3.02	0.78
Water, %	Trace	0.04	0.03	Trace
Sediment, %	0.15	0.11	0.12	1.10
Ash, %	0.024	0.033	0.480	0.045
Residue, % (½ hr. at 400 C)	17.42	24.48	25.06	23.11
Hard asphaltum, %	2.92	4.43	7.25	6.53
Viscosity, SSU at 100 F	114	702	945	287
Cold test, F	—10	2	3	2
Conradson carbon, %	5.49	5.77	10.29	11.87

1—Fuel used in Busch-Sulzer mechanical-injection engine in service 7 years at Union Stock Yards, Chicago.
2—Fuel used in Busch-Sulzer mechanical-injection engine at McMinville, Oregon, during past year.
3—Shell No. 6 fuel used for testing mechanical-injection engines for Maritime Commission.
4—Fuel used for years in successful central station equipped with air-injection engines.

The engine cross section shows the one-piece bedplate which forms two longitudinal girders connected by integral bridges carrying the lower main-bearing half-shells. These are steel, tinned and babbitt lined, and can be rolled out without removing the crankshaft. The bearing cap, babbitt lined, forms the upper main half. Cast iron cylinder supports form spacers between the bedplate and the CI (cast iron) one-piece cylinder block, bored to take seven alloy CI cylinder liners divided into upper and lower barrels.

Water-cooled, individual, CI cylinder heads hold upper barrels; lower barrels mount on underside of block. Bottoms of upper barrels and tops of lowers carry sectional oil-wiper rings which prevent dirty oil and gas from passing into crankcase from above, and clean oil from passing into combustion space from below. The space between upper and lower barrels opens into a chamber in the cylinder block; openings on the block are fitted with glass covers and lights so that piston lubrication may be watched.

Four rotary positive-displacement blowers, arranged in line above the suction header, deliver air to a receiver above, which opens into the cylinders by way of scavenging passages and ports. Scavenging and exhaust ports are so arranged that in downward travel the piston first uncovers the upper tier of scavenging ports; non-return valves cut off communication to the receiver. The piston next uncovers the exhaust ports and later the lower scavenging tier. This produces a moderate amount of supercharging with free exhausting and scavenging, and without risk of "blowback" into the receiver.

Working pistons consist of a forged-steel top carrying rings, oil cooled through telescopic tubes; a plain CI skirt of light symmetrical sections without openings through which wristpin lube oil could escape; and a wristpin housing attached to the top within the skirt, providing a babbitt-lined bearing for the pin. The steel-forged wristpin, hardened and ground on rubbing surfaces, has a flatbottomed recess in the underside to form a seat for the connecting-rod top flange. The wristpin receives lubrication from the piston-cooling supply and not from the crankpin, through the rod. The entire assembly of housing, rod, and pin, can be removed without needing to drive wristpin out of tight fits in piston bosses.

FUEL-INJECTION SYSTEM.

The injection system is a modification of the Bosch type, made by Diesel Equipment Co. Simple differential needle valves, hydraulically operated by fuel-pump pressure, mount on the cylinder heads. Fuel passes to the combustion space through water-cooled atomizing nozzles. Duplicate sets of cams, for running "ahead" or "astern," operate fuel-pump plungers; the crankshaft drives the camshaft through gears. Opening a bypass valve during delivery controls fuel quantity per plunger stroke; excess fuel returns to pump suction. A Woodward governor actuates control gear varying point of bypass opening. Pumps are arranged for manual cutout of fuel delivery to any cylinder, and are connected to a Woodward overspeed governor which interrupts fuel delivery to all cylinders when engine speed exceeds desired maximum. In accordance with Maritime Commission specifications, fuel is heated, centrifuged, and reheated to about 180 degrees F.

A single motor-driven pump serves both piston-cooling and general lubrication systems. Returns from both systems run to a sump built into the ship's structure, but oil from the two does not mix until it leaves the engine, thus minimizing accumulation of hot oil vapors in the crankcase. The pump draws oil from the sump, through a strainer, and discharges back to the engine through a filter and cooler.

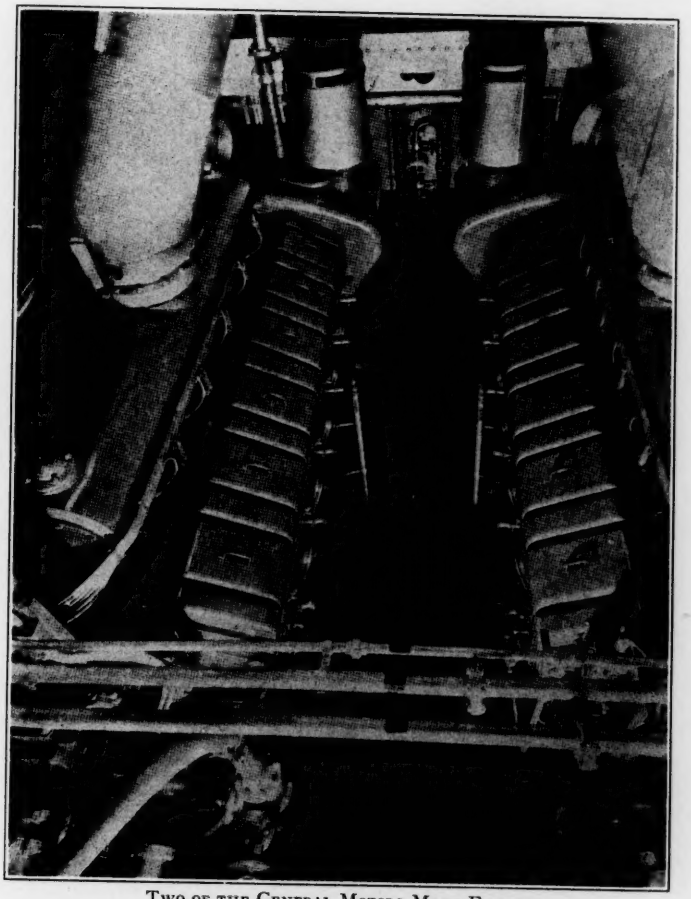
THREE NEW NAVY FLEET TUGS.

The Fleet Tugs, U. S. S. *Navajo*, U. S. S. *Seminole*, U. S. S. *Cherokee*, are driven by Diesel engines. The success of these engines for propulsion in large tugs marks another step in the advance of Diesel main power units for naval use. The paper is by Rex W. Wadman. It is reprinted from the March, 1940, issue of Diesel Progress, published by Diesel Engines, Inc., 2 West 45th Street, New York, N. Y.

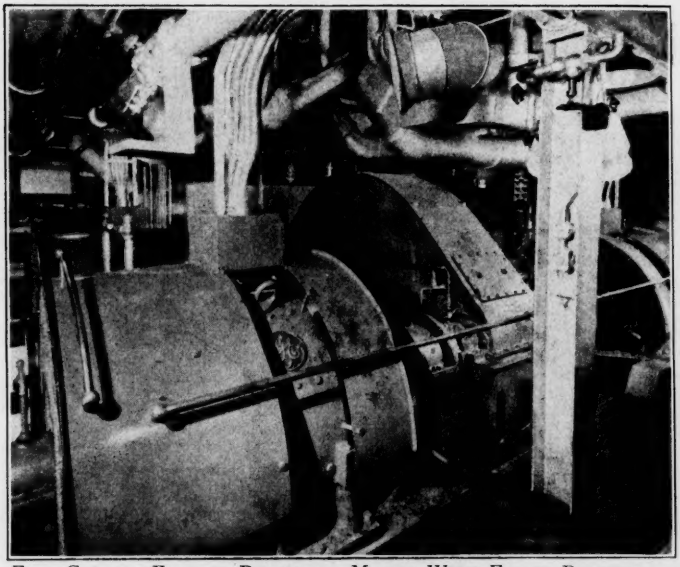
A picture of the U. S. S. *Navajo* is shown in the frontispiece.

Our progressive Navy tore a chunk out of the book of precedents and threw it overboard when it designed the three new Fleet Tugs recently completed at the Staten Island yard of the Bethlehem Steel Company. Selection of Diesel-electric drive for these three big tugs typifies the splendid spirit of progressiveness which permeates our whole Navy in the building up of our first line of defense to be second to none in the world.

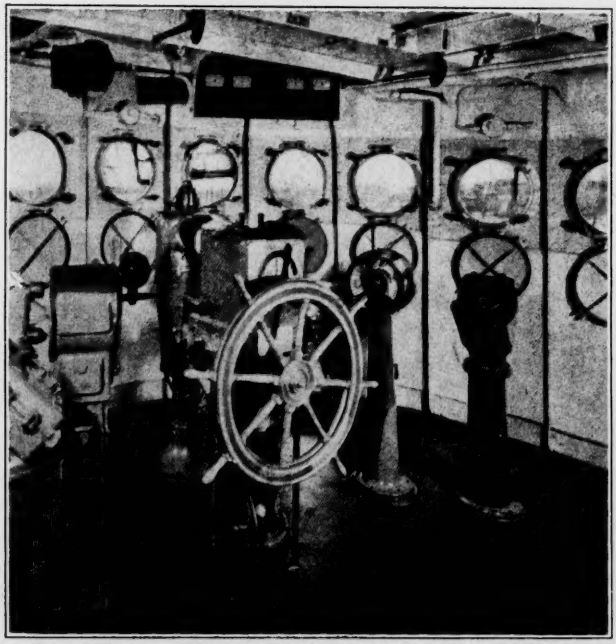
The *Navajo*—the *Seminole*—the *Cherokee*—three fine old American Indian names which have been given to these three Navy Fleet Tugs. Each tug is 205 feet × 38 feet 6 inches × 14 feet 3 inches. Large fuel and lubricating oil capacity provide ample cruising radius. Speed on trials was better than 15 knots.



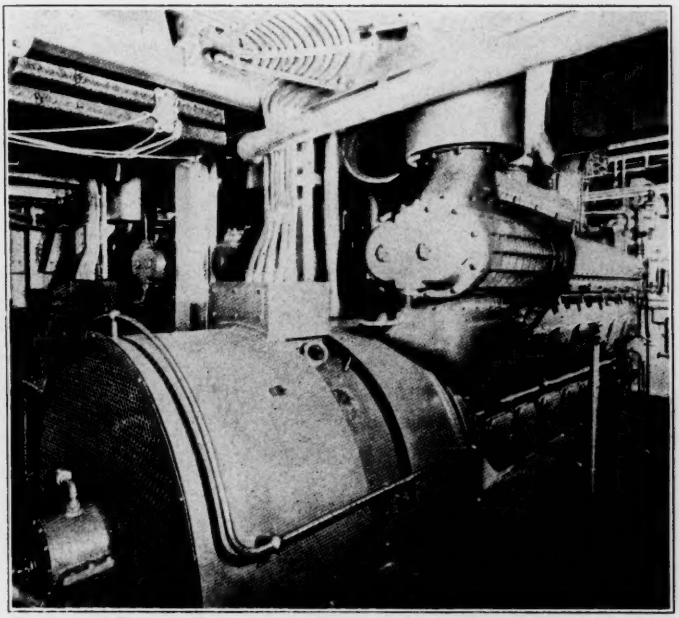
TWO OF THE GENERAL MOTORS MAIN ENGINES.



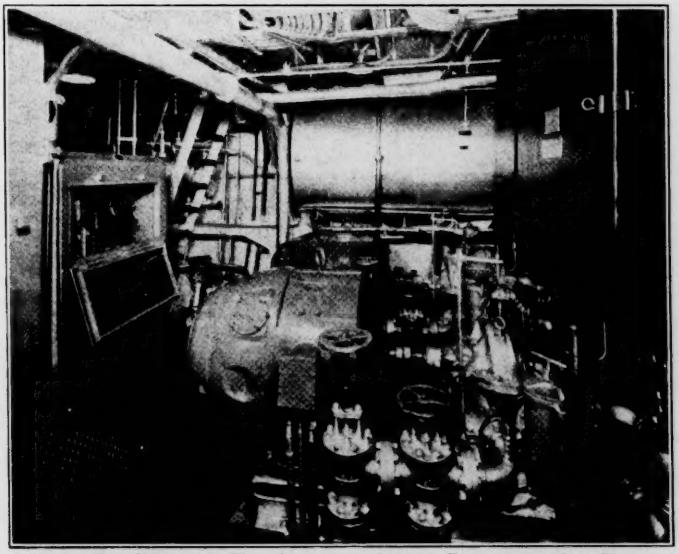
Four General Electric Propulsion Motors With Farrel Birmingham Reduction Gears.



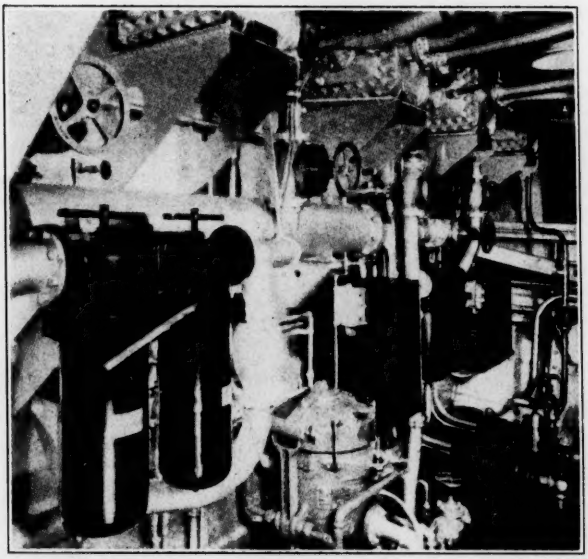
The Pilot House. Weston Tachometers Showing RPM. of Each Main Engine Are Mounted on Ceiling.



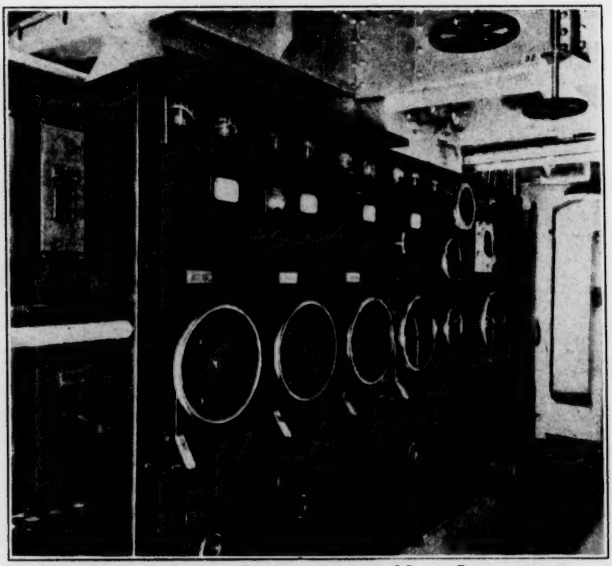
GENERATOR END OF ONE OF THE FOUR GENERAL MOTORS MAIN ENGINES, EACH RATED AT 950 HP.



A PUMP BAY WITH 100 KW. AUXILIARY DIESEL GENERATING SET, ENCASED IN HUSSMAN ACCOUSTIC HOOD AND MOUNTED ON HUSSMAN SPRING BASE, AT LEFT.



Battery of Purolator Lube Oil Filters in Foreground Harrison Heat Exchangers Above and Goulds Hydroil on Lube Line Below.



The Control Panel Section of the Main Switchboard.

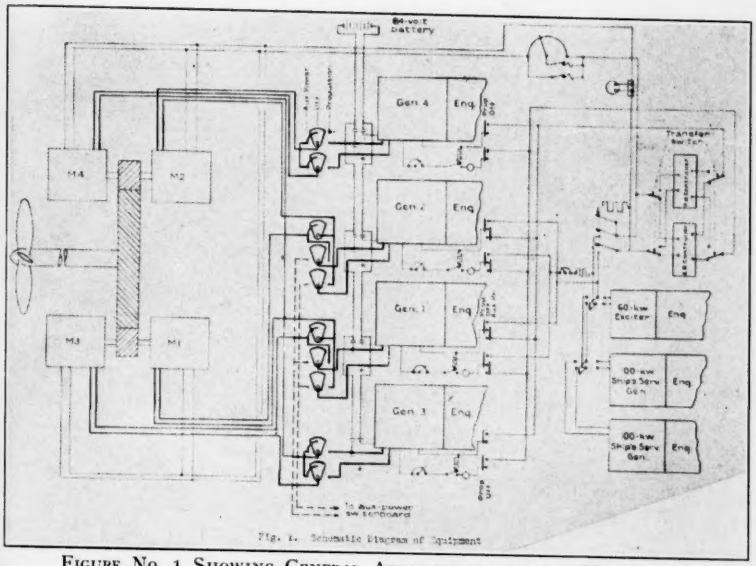


FIGURE NO. 1 SHOWING GENERAL ARRANGEMENT OF THE MACHINERY.

Because these three vessels are essentially first line ships, that is all which can be said at this time about the tugs themselves, but we are permitted to discuss at some length the extremely interesting propulsion equipment, which consists of four model 12-567 General Motors, twelve-cylinder, V-type, two-cycle Diesel engines rated at 950 HP. each at 750 RPM., a total of 3800 HP. of main engines. The auxiliary engine installation consists of two model 233-A General Motors eight-cylinder, four-cycle Diesel engines, each rated at 150 HP. at 1200 RPM., and one model 6-71 General Motors six-cylinder, two-cycle Diesel engine rated at 90 HP. at 1200 RPM.: A total of 390 HP. in auxiliary power, giving a total of 4190 HP. aboard each tug for all purposes.

So, the Navy has pioneered in the adaptation of a multiple Diesel engine installation in these powerful, highly important new additions to the Fleet and the Navy definitely broke with precedent in laying out the engine room for these tugs, not only in selecting Diesels for the main drive, but also in selecting Diesel-electric drive through a reduction gear to one propeller.

In taking this progressive step the Navy has built three ultra-modern Fleet Tugs in which has been incorporated extreme flexibility of operation, extremely low operating costs, plus the minimum amount of space devoted to machinery.

The four main engines with their generators are separated completely from the propulsion motor room by a through bulkhead, a unique feature of design only practical with a Diesel-electric layout.

The propulsion and auxiliary power equipment installed aboard each of the three Diesel-electric tugs includes the following items:

Four main propulsion motors, each 550 volts, single armature type, separately excited from a 120-volt supply.

Four main generators, each 550 volts, separately excited from a 120-volt supply, Diesel-engine driven.

One exciter, rated 60 Kw., 1200 RPM., 120 volts, self-excited, Diesel-engine driven.

Two ship's service generators, each rated 100 Kw., 1200 RPM., 120 volts, self-excited, Diesel-engine driven.

Current for excitation of the four main generators and four main motors is supplied by a separate General Motors, Model 6-71 two-cycle Diesel engine, direct-connected to a 60 Kw. exciter-generator. Emergency excitation may be supplied by either of the two 100 Kw. General Motors Diesel-driven ship's service generator sets. The 60 Kw. engine-driven exciter supplies power to the excitation bus for the fields of the main motors and generators. The 100 Kw. ship's service generators can also supply power to the excitation bus, but the switches are arranged so that only one of these machines can be connected to the excitation bus at one time. The 100 Kw. generators also supply the general ship's service power and lighting circuit. Either GEN 1 or GEN 2 may be connected to the auxiliary-power bus by throwing the main switch to the AUX. P. position. The main switches of these two main generators are interlocked so that only one can be thrown to the AUX. P. position at once.

The arrangement of the equipment is shown in Figure 1. Power for the propulsion motors is furnished by the main generators. The motors drive the propeller through reduction gears manufactured by the Farrel-Birmingham Company. The propulsion gear unit is of the two bearing, single reduction type, fitted throughout with Timken roller bearings and provided with a self-contained lubricating system.

The speed of the vessel can be controlled from the pilot house or from the engine room by varying the strength of the generator field and/or the speed of the driving engines. It is also possible to use either one or two

of the main generating units to supply power for extra heavy auxiliary loads, such as the electrically-operated towing engine.

The main propulsion control consists of four generator panels and one motor panel. Each one of the main generator panels contains a voltmeter, generator field ammeter, and engine speed indicator, together with a hand wheel which connects the corresponding generator for starting or propulsion. In addition, the hand wheels on two of these panels control the set-up for providing the power for the extra heavy auxiliary loads. The motor panel contains the following instruments: motor ammeter, motor field ammeter, excitation voltmeter, motor and ground detector, voltmeter, master shaft revolution indicator, engine order telegraph indicator. It also contains the main control operating hand wheel, transfer control hand wheel, and motor field rheostat hand wheel.

After all engines are connected to the propulsion bus, the boat is operated from full speed ahead to full speed astern by operating the one control operating hand wheel on the motor panel, or, if desired, the transfer switch may be set for operating from the pedestal control in the pilot house.

It can be seen that complete flexibility is obtained in that the operator has control of both the number of engines used for propulsion and the engine speed. This flexibility permits the ship to be operated even above half speed by the use of but one main engine at very little over the idling speed of the engine.

General Electric Company supplied all the main generators, propulsion motors and electric auxiliary power equipment, and Woodward governors are on all main and auxiliary Diesels. Harrison heat exchangers are used throughout all three tugs for the closed cooling water system and for the lube oil lines. There is a Goulds Hydroil on the lube line and the fuel line in each tug. Purolator filters are installed on lube lines of all main engines. An interesting feature of the three auxiliary engines is that they all are mounted on Hussman Spring Mountings to eliminate vibration and each auxiliary engine is encased in a Hussman Accoustic Hood to segregate engine noise from the rest of the ship when nothing but auxiliary engines are in operation or needed, thus greatly contributing to the comfort of the officers and crew. Engineering Specialties supplied the Vortex spark arresting silencers and there is a Weston tachometer for each main engine on the switchboard and mounted in the pilot house.

Again, congratulations to our United States Navy for progressive thinking and acting in building these three fine Diesel Electric Fleet Tugs.

TESTING GEAR-WHEEL MATERIAL.

A new apparatus developed by Brown Boveri for the testing of gear-wheel material is described in this article. Some results of tests are given. It is shown that the direction of sliding movement plays a preponderant part in the durability of the material. The article is by A. Meldahl. It was published in the October, 1939, issue of the Brown Boveri Review, Baden, Switzerland, obtainable through A. Francke A. G., Berne and Rouge & Cie., Lausanne, Switzerland.

Brown Boveri resumed investigations into the running properties of the faces of gear-wheel teeth, in conjunction with orders received for gear wheels of very big dimensions (see The Brown Boveri Review of January-February, 1938, page 38). It is of great importance in such tests that

the test pieces should be of as simple design as possible, so that they can be manufactured with ease and precision, while not being too expensive. In accordance therewith, simple cylindrical test bodies were used; these roll under load, one on the other, in a testing apparatus.

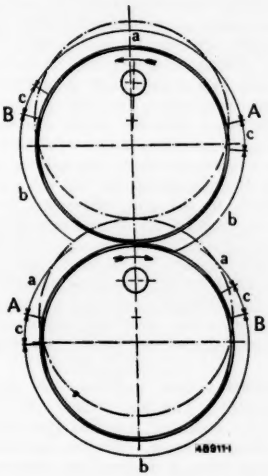


FIGURE 1.—MOVEMENT OF THE TESTING PIECES IN THE BROWN BOVERI TESTING APPARATUS FOR GEAR-WHEEL MATERIAL.

A. Rolling Point, *Driving* Face.
B. Rolling Point, *Driven* Face.
a. Dedendum Arc.
b. Addendum Arc.
c. Rolling Zone.

Now, in order to reproduce faithfully the conditions pertaining to teeth faces, it is necessary to generate a sliding movement between the rollers. It is not desirable, to this end, simply to cause the shafts to rotate at different speeds. However, it is possible to create a sliding movement even when both shafts have the same number of revolutions, by making the test cylinders of different sizes or by coupling the shafts through eccentric involute gear wheels. The latter is the solution generally adopted in the present tests. It allows of the faithful reproduction on the periphery of one single test cylinder of all the movements occurring between two tooth faces and thus of determining the behavior of the material by means of a few tests only.

If the two shafts are coupled by eccentric involute gear wheels, one test piece leads angularly on the other during about one half revolution and lags by the same amount during the next half revolution. In two points only the sliding velocity vanishes, these two rolling points A and B divide the periphery into two parts of unequal length, a and b. These two parts have

different sliding conditions. On the shorter length a, the rolling movement and the sliding movement are in opposite directions while on the longer length b they are in the same direction. On the other hand, the sliding movement, that is the relative movement of the other body, is always from rolling point A towards rolling point B.

If the sliding and rolling movements of the teeth of a gear wheel are studied, the following results are obtained:— the rolling movement on a *driving tooth* is *from root towards tip* while on a *driven tooth* it is from *tip towards root*. The sliding on a driving tooth is always directed away from the *rolling point* and on a *driven* tooth it is directed *towards the rolling point*. Therefore, on the *addendum*, the rolling and sliding motions are in the *same direction* while being *opposite* on the *dedendum*.

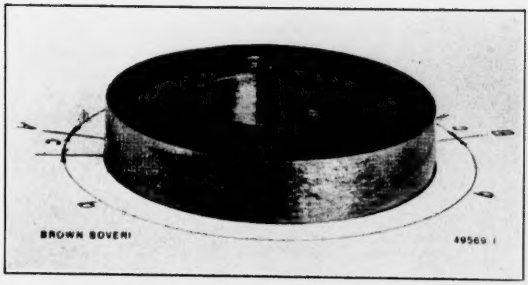
The comparison between test pieces and tooth faces leads to the following results:— rolling point A corresponds to the rolling point of a *driving* tooth face while rolling point B corresponds to that of a *driven* tooth face. The shorter arc a corresponds to the dedendum and the longer arc b to the addendum. Thus, all the conditions of movement to be found in a gear drive are faithfully reproduced on the periphery of the test cylinder.

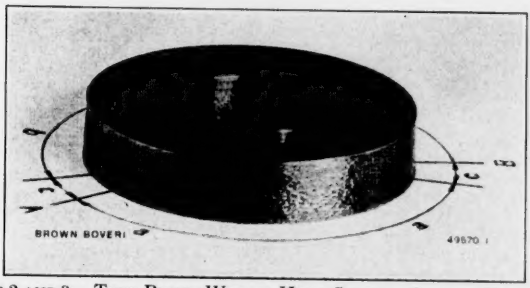
Starting from the rolling points, there are small *rolling zones* c created by the clearance of the teeth of the coupling gear wheels. On the rolling point the direction of the friction force changes and the test rollers then control the movement themselves until the clearance in the teeth of the coupling wheels has caught up and the opposite tooth faces take over control (see Appendix).

The new Brown Boveri testing machine for gear materials has been designed on these principles. The test rollers are mounted on two heavy shafts which also carry the eccentric coupling gear wheels. The two shafts lie in half bearings in a hinged frame. The two cylinders are pressed against each other, with whatever force is desired, by means of a lever loaded with weights. The position of the test roller on the shaft is determined by a pin so that its position thereon cannot change and modify that of the rolling point. An arrow is stamped into the test roller to mark the sense of rotation during the test. The test rollers are shrunk on to the shafts. To remove them, test roller and shaft are heated up in an oil bath whereupon the hollow shaft is loosened by passing a jet of cold water through the bore. Test rollers, bearings and gear wheels are amply lubricated by an oil pump with separate drive. In order that the oil be maintained at constant temperature, a refrigerating coil and a heating coil are lodged in the sump, the heating coil being necessary for tests with heavy oils.

Generally, the load on the test rollers during the tests carried out was 880 pounds or about 1500 pounds per inch, which, with the dimensions chosen and according to the Hertz formula, corresponds to a pressure of 45 tons per square inch, which is considerably more than is admissible for gear teeth which have not been hardened. The maximum specific sliding (ratio of sliding to rolling velocity) is -0.67 for the arc a (dedendum) and $+0.40$ for the arc b (addendum). The average rolling velocity at full speed is about 13 feet per second; it was, unfortunately, impossible to go higher on account of the coupling gear wheels. The angular acceleration of the upper shaft then attained up to 6000 s^{-2}, the coupling gear wheels and the shafts being, consequently, very highly stressed. Ordinary turbine oil was generally used as a lubricant, transformer oil and cylinder oil being used, however, in some cases.

The illustrations show the appearance of some test rollers after the tests. Details on the conditions of the tests are given beneath the illustrations.





FIGURES 2 AND 3.—TEST PIECES WHICH HAVE SEIZED. SEIZING TAKES PLACE IN THE MIDDLE OF THE DEDENDUM ARC a, OR THE ADDENDUM ARC b, RESPECTIVELY, WHERE SPECIFIC SLIDING IS MAXIMUM.

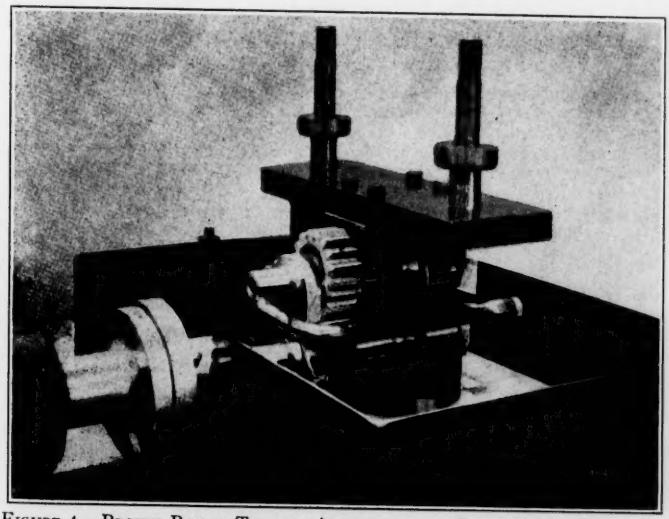


FIGURE 4.—BROWN BOVERI TESTING APPARATUS FOR GEAR-WHEEL MATERIAL.

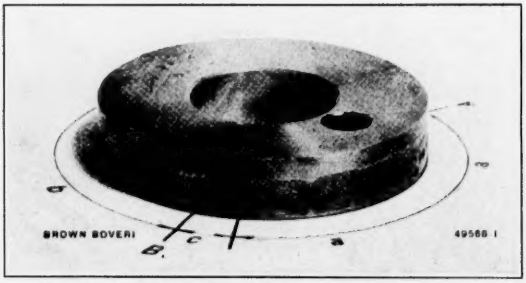


FIGURE 5.—TEST PIECE WITH PITS. PITTING BEGINS SHORTLY AFTER THE ROLLING POINT B.
Big Pits Are Caused by Lubrication With Transformer Oil.

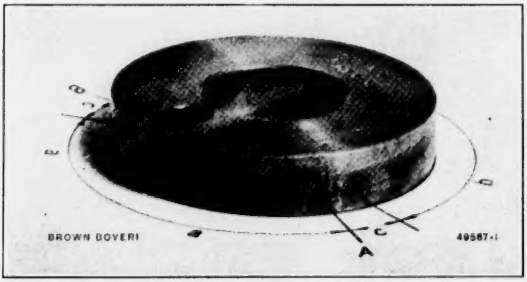


FIGURE 6.—TEST PIECE WITH PITS. PITTING CEASES EXACTLY ON ROLLING POINT A.
Big Pits Are Caused by Lubrication With Transformer Oil.

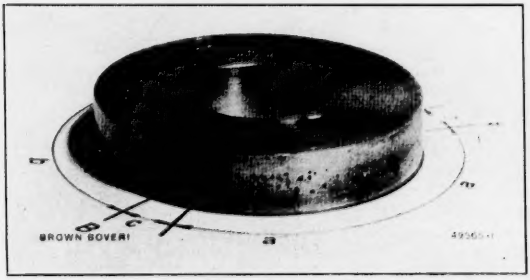


FIGURE 7.—TEST PIECE WITH PITS. PITTING BEGINS JUST AFTER ROLLING POINT B.
Small Pits Caused by Lubrication With Cylinder Oil.

At the present stage of investigations, the results can be summarized as follows:

The test bodies can be damaged in two quite different ways—either by *seizing* (Figures 2 and 3) or by *pitting* (Figures 5-12).

Damage by seizing occurs chiefly with soft steels. It generally occurs after a relatively short time, often, even, after a quarter of an hour. The test bodies seize at the point where the specific sliding is the greatest, that is to say midway between the rolling points. *Both* test rollers are equally attacked, this though the damage on the dedendum arc a is naturally shorter than that on the addendum arc b.

It is possible to prevent seizing by lubricating with cylinder oil, while thin transformer oil increases the risk of seizing. If the test rollers are run in under loads, gradually increased, it is generally possible to prevent seizing altogether.

With both rollers of the same material, the addendum arc b of the lower roller generally seizes first, as does the dedendum arc a of the upper roller, because here the average speed is the lowest. With different materials, the dedendum arc a of the softest material seizes first and thus destroys the addendum arc b of the harder material.

All this data is valid for a maximum specific sliding of -0.67 on the dedendum arc a and $+0.40$ on the addendum arc b. With preliminary tests with lower specific sliding, namely -0.46 on the dedendum arc a and $+0.32$ on the addendum arc b, there was never a case of seizing with test bodies of the same material.

This shows that, in the case of seizing, it is a question of lubrication and that the seizing is determined by a given magnitude of the specific sliding. These results conform to those of practical experience with gears. Seizing is chiefly encountered with new gears with profile displacement and at low speeds, especially when a relatively soft material is used. By careful running in under increasing load and by using heavy lubricating oils—especially during running in—seizing can be eliminated.

Pitting.—Contrary to seizing, pitting only appears after a longer testing time. Small particles break out of the roller surface, the appearance of a typical pit is that of a sector of a circle with an angle of 90 degrees to 120 degrees. The fracture begins in the center of a sector in formation and penetrates from here at about 15 degrees beneath the surface. On the surface, all that is to be seen, at first, are the two radii as very fine cracks, until, finally the crack has advanced so far that the whole wedge now loosened—probably under the pressure of the lubricating oil forced in—is lifted out of the surface and breaks off in circular shape.

The surface of the fracture is shell like and it can be seen that the fracture has advanced progressively as takes place in a fatigue fracture. It is probable that it really is a corrosion fatigue, because many test pieces, after a time, show a thin reddish coating, which adheres very firmly to the surface and which probably consists of iron oxides.

The tests show quite clearly that, contrary to seizing, pitting appears *exclusively on one of the two test bodies.* As is seen in all the illustrations, the *pitting only appears on the dedendum arc a* whereas the addendum arc b is always unaffected.

The limitation of pitting to the dedendum arc a is very sharply defined indeed. The pitting stops abruptly at rolling point A, the limit is sharply defined and coincides with the theoretically determined rolling point, to one-hundredth of an inch. The beginning of pitting at rolling point B takes place a short distance after the rolling point in the rolling zone, it should

be remembered, however, that quite small changes in the clearance of the teeth or distance between the shafts influence the width of the rolling zone considerably, so that the point in which the sliding movement begins cannot be determined with the same certitude as can rolling point A, at which the sliding movement ceases.

Thus, the tests show that the pitting is furthered very much by sliding, but only when the sliding movement is *counter* to the direction of the rolling movement. The proclivity to pitting is increased to an astonishing degree and the illustrations show some exceptional examples. In one case with a roller of heat-treated chromium-nickel steel having a yield point of abt. 48 tons per square inch much pitting appeared on the dedendum arc a, while on the other roller of carbon steel with a yield point of abt. 29 tons per square inch no pitting occurred at all on the addendum arc b. On another roller the width of the addendum arc b was reduced to less than half, and the linear pressure, thus, more than doubled. Despite this, the addendum arc b remained intact while the dedendum arc a showed marked pitting.

The influence of the viscosity of the lubricating oil is also of interest—the more fluid the lubricating oil the bigger the pit. With transformer oil, big pits of 5 millimeters diameter are formed, while with cylinder oil the pits are hardly bigger than a pin head. This seems to confirm the view that the lubricating oil forced into the fatigue cracks helps pit formations and finally lifts the loose material from the surface. A heavy cylinder oil will be more likely to detach the material loosened from the lower layer and thus to complete the formation of a pit than a thin transformer oil which can easily escape from the narrowest cracks.

In order to investigate more completely the resistant character of the addendum arc, further tests were carried out with somewhat different apparatus. As said before, it is possible to generate a sliding movement of the two test rollers, one on the other, by imparting uniform running to the shafts but by using test rollers of different sizes. The sliding movement is then the same at every point. With the dimensions chosen, here, the specific sliding on the smaller test piece was -0.29 and on the bigger one $+0.22$.

The result of these tests was the following:

The smaller test pieces were made of carbon steel and of nickel steel with a yield point of 29 and 25 tons per square inch, respectively. The bigger test piece was of chromium-nickel steel with a yield point of 48 tons per square inch. After three to six hours, pits appeared on the smaller roller. The linear pressure was 1500 pounds per inch in this test, the speed being 1000 RPM.

The contrary arrangement, that is to say with smaller test pieces of chromium-nickel steel having a yield point of 48 tons per square inch and the bigger ones of carbon steel and nickel steel having yield points of 29 and 25 tons per square inch, respectively, stood up to 100 running hours without any pitting at all. In order to force deterioration of the addendum arc, the surface of the bigger cylinder was turned down to about half its width and the load increased as much as the safety of the bearings would permit. The linear load was then 3500 pounds per inch, the stressing, according to the Hertz formula, 70 tons per square inch. After about 100 hours of running, in one case the smaller chromium-nickel roller broke down; in spite of having a yield point of 48 tons per square inch, it could not stand up to the bigger nickel-steel cylinder having a yield point of only 25 tons per square inch. In a second case a bigger roller of carbon steel with a yield point of 29 tons per square inch running at 1000 RPM. stood up to this enormous load for 150 hours without showing any visible signs of fatigue, whereas a smaller

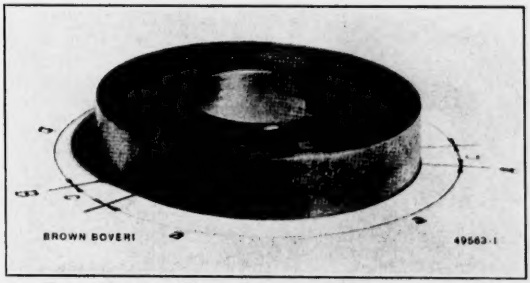


FIGURE 8.—TEST PIECE OF HARD CHROMIUM-NICKEL STEEL WITH A YIELD POINT OF 48 TONS PER SQUARE INCH PITTING ON THE DEDENDUM ARC a.

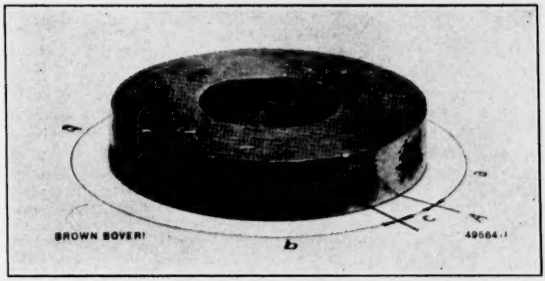


FIGURE 9.—TEST PIECE OF CARBON STEEL WITH A YIELD POINT OF 29 TONS PER SQUARE INCH. PITTING ON DEDENDUM ARC a CEASES EXACTLY ON ROLLING POINT A.

These Two Test Pieces Ran Together. The Dedendum Arc a of the *Hard* Test Piece Figure 8 Shows Heavy Pitting, While the Addendum Arc b of the *Soft* Test Piece Figure 9 Is Not Attacked at All.

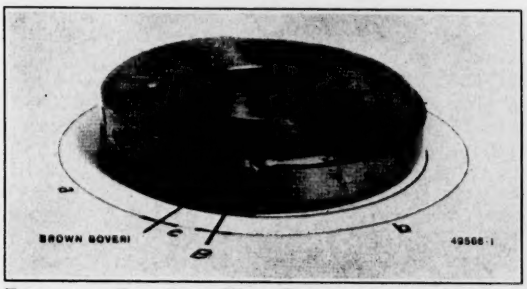


FIGURE 10.—TEST PIECE WITH REDUCED ADDENDUM ARC b.

Although the Addendum Arc b Has Twice the Linear Pressure, It Has Not Been Attacked, While the Dedendum Arc a, With Double Width, Shows Heavy Pitting Beginning Just After Rolling Point B.

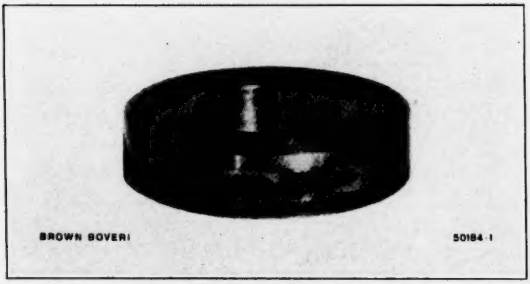


FIGURE 11.—SMALLER TEST PIECE OF CHROMIUM-NICKEL STEEL WITH A YIELD POINT OF 48 TONS PER SQUARE INCH. HEAVY PITTING.

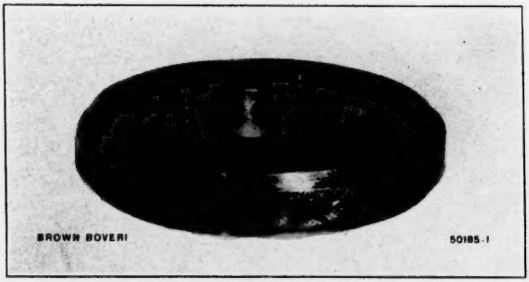


FIGURE 12.—BIGGER TEST PIECE OF NICKEL STEEL WITH A YIELD POINT OF 25 TONS PER SQUARE INCH. NO PITTING.

These Two Test Pieces Ran for 150 Hours at 1000 RPM., Load 500 Kilograms, Corresponding to a Stress of 70 Tons per Square Inch According to the Hertz Formula. The *Small* Test Piece of Hard Steel Is Much Damaged, While the Big One of Soft Steel Is Not Damaged at All, Although Plastically Deformed.

cylinder of the same material, as already mentioned, broke down after a few hours running at only 1500 pounds per inch linear load.

No statement can be made, on the basis of results attained so far, as to what a material will really support on the addendum arc. The tests, however, are being pursued along these lines. We are at a loss, up till today, to furnish a plausible explanation of why the same material should support such exceptionally different stressings according to the *direction* of the sliding movement. Perhaps subsequent tests will give the answer to this problem.

APPENDIX.

Kinematics of New Test Apparatus.

Two involute gear wheels in mesh can be considered, kinematically, as being two pulleys the diameters of which are equal to the diameters of the fundamental circle of the involute gear wheels and which are connected by a thin crossed belt.

In the drawing above, I and II are the centers of the two shafts, I′ and II′ are the centers of the coupling gear wheels. B is the constant distance between the two shafts which is determined by the diameter of the test pieces. A is the variable distance between the gear wheel centers. The eccentricity of both gear wheels is equal to e; φ_1 and φ_2 are the angles of revolution of the shafts, α the pressure angle and β the angle between the two center lines of the shafts and centers of the gear wheels, respectively. Two meshed wheels can, speaking generally, perform the following distinct movements:

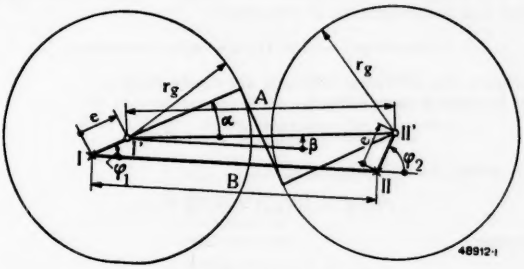


FIGURE 13.—DRIVE BY ECCENTRIC INVOLUTE GEAR WHEELS.

I, II. Center Points of Shafts (Immovable).
I′, II′. Center Points of Involute Gear Wheels.
A. Distance of Centers of I′-II′ of Involute Gear Wheels (Variable).
B. Distance of Centers of Shafts I-II (Constant).
e. Eccentricity.
r_g. Radius of Fundamental Circle of the Involute Gear Wheels.
α. Pressure Angle.
β. Angle Between A and B.
φ_1. Angle of Rotation of Wheel I.
φ_2. Angle of Rotation of Wheel II.

1. Wheel 1 may rotate round I′.
2. Wheel 2 may rotate round II′.
3. Wheel 2 may rotate round wheel 1 or wheel 1 round wheel 2 at a constant distance A, each wheel retaining its direction in space. It, therefore, goes through a purely parallel movement.
4. The distance A may vary, that is to say the pressure angle α may change.

We will now investigate what variations in the length of the imaginary belt are caused by these movements.

1. The rotation of gear wheel 1 round I′ causes the belt to lengthen by

$$+ r_g \cdot \varphi_1.$$

2. The rotation of gear wheel 2 round II′ causes the belt to lengthen by

$$+ r_g \cdot \varphi_2.$$

3. The mutual rotary movement of the gear wheels causes the belt to shorten by

$$- 2r_g \cdot \beta.$$

4. Change in the distance A of the centers, i.e., the increase in the pressure angle, causes the belt to lengthen by

$$+ 2r_g (+ \operatorname{tg} \alpha - \alpha).$$

This is valid for the pressure line shown in the drawing, that is to say, for pressure of the right tooth face. For pressure of the left face, the sign of α changes. The length of the belt must not change during the movement, which leads to the following equation of movement

$$\tfrac{1}{2} (\varphi_1 + \varphi_2) - \beta \pm (\operatorname{tg} \alpha - \alpha) = \text{constant}. \qquad (1)$$

Furthermore, the following relations are easily deduced:
For the horizontal coordinates:

$$e (\cos \varphi_1 - \cos \varphi_2) + A \cos \beta = B. \qquad (2)$$

For the vertical coordinates:

$$e (\sin \varphi_1 - \sin \varphi_2) + A \sin \beta = 0. \qquad (3)$$

And, further,

$$A \cos \alpha = 2 r_g = B \cos \alpha_0. \qquad (4)$$

These four equations determine the movement of both test pieces; they can only be computed by trial and error. The constant of equation (1) must be determined from the initial condition.

If the equations are differentiated and if A, A′, α' and β' are then eliminated, the following equation is obtained:

$$\varphi_1' \left(1 + \frac{e}{r_g} \cos (\varphi_1 - \alpha - \beta)\right) + \varphi_2' \left(1 - \frac{e}{r_g} \cos (\varphi_2 - \alpha - \beta)\right) = 0 \qquad \ldots (5)$$

This equation might have been deduced from the drawing which serves as a check.

Equation (5) allows of determining the expression for specific sliding

$$\frac{\phi_1' + \phi_2'}{\phi_1'} = \frac{e}{r_g} \cdot \frac{\cos(\phi_1 - a - \beta) + \cos(\phi_2 - a - \beta)}{1 - \frac{e}{r_g}\cos(\phi_2 - a - \beta)} \qquad (6)$$

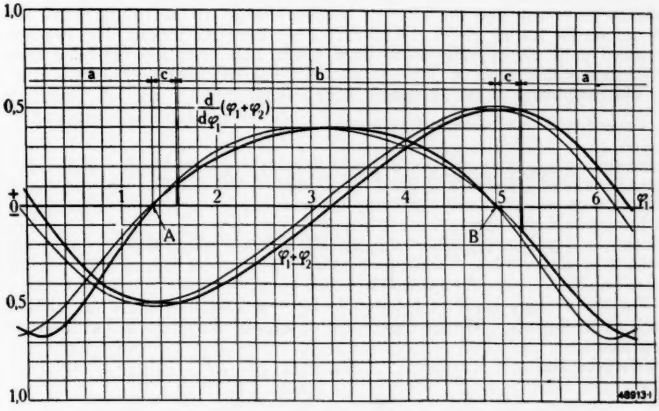


Fig. 14. — Movement of the upper test piece in the Brown Boveri testing apparatus for gear wheel material.

A. Rolling point of **driving** face.
B. Rölling point of **driven** face.
a. Dedendum arc.
b. Addendum arc.
c. Rolling zone.
φ_1. Angle of rotation of lower test piece (constant speed).
$\varphi_1 + \varphi_2$. Angular deviation of upper test piece.
$\frac{d}{d\varphi_1}(\varphi_1 + \varphi_2)$. Specific sliding of upper test piece.

The double curves correspond to drives by the right and by the left face of the eccentric involute gear wheels respectively. The distance between the curves is determined by the tooth clearance. In the rolling zone c, the drive changes from one face to the other.

The rolling points are determined by the equation (6). Sliding ceases when

$$\cos(\varphi_1 - \alpha - \beta) = -\cos(\varphi_2 - \alpha - \beta),$$

that is to say when

$$\varphi_1 - \varphi_2 = \pi;\ 3\pi;\ 5\pi, \ldots \text{ etc.}$$

This determines the theoretical rolling points. As, however, there must be a certain clearance between the gear wheels, rolling zones of definite width appear in conjunction with the theoretical rolling points. The direction of the sliding movement and the force of friction change on the rolling point and, therefore, the testing pieces roll on each other after the rolling point has been reached until the opposite tooth faces touch, that is to say until the tooth clearance is taken up.

The calculation of the movement is as follows:

According to well-known formulae:

$$\sin\varphi_1 - \sin\phi_2 = 2\sin\frac{\phi_1 - \phi_2}{2}\cos\frac{\phi_1 + \phi_2}{2}$$

$$\cos\varphi_1 - \cos\phi_2 = -2\sin\frac{\phi_1 - \phi_2}{2}\sin\frac{\phi_1 + \phi_2}{2}$$

For the average angle of rotation $\frac{1}{2}(\varphi_1 - \varphi_2)$ uniform steps are chosen, such as 30 degrees. Then a magnitude of angular deviation $\frac{1}{2}(\varphi_1 + \varphi_2)$ is assumed.

We then find

$$\operatorname{tg}\beta = -\frac{2\frac{e}{B}\sin\frac{\phi_1 - \phi_2}{2}\cos\frac{\phi_1 + \phi_2}{2}}{1 + 2\frac{e}{B}\sin\frac{\phi_1 - \phi_2}{2}\sin\frac{\phi_1 + \phi_2}{2}}$$

$$\cos\alpha = \frac{\cos\beta\cos\alpha}{1 + 2\frac{e}{B}\sin\frac{\phi_1 - \phi_2}{2}\sin\frac{\phi_1 + \phi_2}{2}}$$

The values of β and α, thus determined are inserted in equation (1). If the value $\frac{1}{2}(\varphi_1 + \varphi_2)$ was correctly chosen, equation (1) is satisfied, otherwise a residual error Δ is found. In order to eliminate this error a correction $\delta\frac{1}{2}(\varphi_1 + \varphi_2)$ must be made, which is determined from the equation

$$\delta\tfrac{1}{2}(\varphi_1 + \varphi_2) = \qquad (7)$$

$$\frac{\Lambda}{1 - \frac{e}{r_g}\sin\left(\frac{\phi_1 - \phi_2}{2}\right)\sin\left(\frac{\phi_1 + \phi_2}{2} - \alpha - \beta\right)}$$

The calculation converges rapidly.

The curve sheet given here is calculated for the following data for the coupling gear wheels:

Number of teeth	17
Modulus (diametrical pitch)	4.5 millimeters
Pressure angle	20 degrees
Distance of centers at zero clearance	76.6 millimeters
Eccentricity	10 millimeters
Distance between shaft centers	80 millimeters

MISCELLANEA OF INTEREST TO NAVAL ENGINEERS.

MEET THE MILLIBAR.—The Technology Review, February, 1940, published for the Alumni Association of the Massachusetts Institute of Technology, Cambridge, Mass.

The new year has brought a new word and a new idea to the American vocabulary to add further confusion, at least temporarily, to the already perplexed mind of the amateur weather prophet. We sat down to breakfast and the morning paper just after 1940 had become established, and turned to the weather report, only to find our old barometric pressures disguised by aliases. The report fairly bristled with "millibars," and there were a short explanation and a table telling us that 0.089 inches of mercury would henceforth be called three millibars.

As disturbed as anyone would be over the death of a dear friend, at the first opportunity we pressed our associates on the Institute's meteorological staff for an explanation. We were told: "A millibar is a unit of pressure equivalent to 1000 dynes per square centimeter." That sounded like a nice round number, and we were sure that it must be good for something. "Or it is a thousandth of a bar, which, is, of course, a million dynes per square centimeter," the expert continued. "Of course, there's also a chemist's bar, which is equal to one dyne per square centimeter; but that is only a millionth part of the meteorologist's bar and has nothing to do with weather."

Trying to be tactful, we asked what that meant in plain, nontechnical English. "Well, take normal atmospheric pressure, for instance," the weather expert responded. "That is 14.66 pounds per square inch, or 29.92 inches of mercury, or 33.9 feet of water, or 1013.25 millibars, or, of course, 1.01325 bars."

Thoroughly confused, we changed our course and asked for the origin of the idea and its application. As the whole explanation pieced itself together, we began to see that it was completely logical and really not so complicated as it appeared at first. We learned that the Norwegian meteorologist, V. Bjerknes, introduced the term many years ago and that it has long been in use in Europe. The British adopted the term in 1914 for general use and it has been used widely even in this country by meteorologists concerned with the study of weather in the upper air—in the troposphere, to be technical.

On January 2, the millibar was, in effect, brought down to earth in this country by the United States Weather Bureau, of which our own Carl-Gustaf A. Rossby is assistant chief. It is now recognized as the standard of measurement of pressures at the surface of the earth, as well as in the upper air.

Thus American usage now corresponds with a workable international meteorological standard. The adopted unit matches all other metric units in air-mass calculations for predicting weather. Furthermore, the millibar

is much more logical, since it is strictly a unit of pressure. As the shock of losing the mercurial standard subsides, we agree that it was rather absurd to designate pressure in terms of "inches" of a column of mercury when we really were dealing with force per unit of area. Moreover, measurements in inches always had to be converted into millibars before the measurements could be used in making thermodynamical calculations.

Meterologists and scientists in general, who must deal with data recorded in many parts of the world, regard the adoption of the millibar in the United States merely as one important step in the right direction—toward the adoption of the metric system for all units of measurement, including, eventually, the all-important shift from Fahrenheit to centigrade in temperature measurements.

Admitting that it will at first be inconvenient, even annoying, for the millions of amateur students of the weather, our meteorologists are convinced that we shall soon become accustomed to thinking in terms of millibars just as well as we now think in terms of inches. As a special aid in making this transition, they suggest that we fix the number 1013 in our minds as the normal atmospheric pressure in millibars and remember that a tenth of an inch by the old standard is approximately three millibars by the new (actually, 0.1 inch equals 3.37 millibars). So be it!

THE RHUMBATRON.—The Industrial Bulletin of Arthur D. Little, Inc., Cambridge, Mass., February, 1940.

Since the birth of popular radio the tendency has been toward use of continually higher frequencies and shorter wavelengths. Aside from the experimental value to the amateur, the shorter waves are used in television, facsimile transmission, and situations in which directional beams are useful. During the past year announcements bordering upon the sensational have shown the way to effective operation with wavelengths previously outside the practical range. Commercial broadcasting waves are in the order of several hundred meters, television broadcasts are being made at 4 to 8 meters and amateurs transmit messages using down to three-quarters meters. Now operating wavelengths measured in centimeters are in the offing, and, what is more important, means are now, for the first time, available for having these carry a really sizable output of power.

Practical use of ultra-high frequencies generated by standard vacuum tubes has been limited by the fact that, as the frequency increases, the size of the tube, and with it its capacity to transmit electrical energy, must be decreased. Successful escape from this limitation has been based on replacement of the standard tube by a new device called the klystron, the vital part of which has been dubbed the rhumbatron by an imaginative physicist. Curiously, successful developments in this field were announced almost simultaneously during the past year by three laboratories—those of Stanford University, RCA, and General Electric.

The newer wave generators differ from the usual hot-filament tubes in that they depend upon modulations impressed upon a beam of electrons. Electrons are expelled from a heated cathode and are focused through the "hole" of a doughnut-shaped chamber called a rhumbatron. In its passage the smooth electron flow is broken up into bunches of electrons as a result of the natural electrical resonating period of the chamber. The stream then passes through a second rhumbatron which is electrically linked with the first, so that the oscillations of the two are mutually strengthened. It is

this energy pulsation in the second rhumbatron which produces the current oscillations of the circuit. Frequency is determined predominantly by the dimensions of the apparatus.

An output of 500 watts at 40 centimeters, and of around 100 at 10 centimeters, is reported. Wavelengths are often discussed glibly and handled familiarly, but waves in the order of centimeters represent oscillation frequencies so high as to challenge the imagination. Ten-centimeter waves, for example, correspond to an alternating current frequency of three billion cycles per second. Grandfather's clock ticked off the seconds during a lifetime of 90 years to a total of about 2,840,000,000 oscillations of its pendulum. It would be necessary to crowd all of these movements into the space of one second in order to have the pendulum frequency equal that of the current in the rhumbatron circuit.

To what uses such high-frequency power may be put perhaps cannot be fully predicted. Hitherto the necessary power generator has been lacking, so that little has been done by way of application in the arts and sciences. Where extremely short waves have already proven their value, the power need is ready to be filled. An instrument-landing control system uses short waves as an airplane glide path, and this use for the klystron has been successfully tested. It has been stated that in the waveband range from one centimeter to one meter there is room for some 270,000 non-interfering signal frequencies. Although the distance attainable in this range would be limited to the horizon, a transmitter located on a tall building in the center of a city and using the one centimeter to one meter band could reach as many receivers as there are telephones in a city of about 1,000,000 population. The waves are, however, deflected and disturbed by buildings and other obstacles. The extremely short waves may be highly focused with a minimum of equipment, thus concentrating their power. As in the recently announced absolute altimeter for airplanes, these focused beams may be useful for the location of large obstacles in both aerial and marine navigation.

The new oscillators are not seriously restricted in size and ultra-high-frequency power measured in kilowatts may become an actuality. This possibility re-opens the question of applying induction effects to uses previously excluded because sufficient power could not be obtained at the requisite high frequencies. Many materials normally considered to be electrically non-conducting respond to high-frequency electric fields; there is an absorption of energy with a direct conversion to heat within the body. At the present time, applications of this sort are being made up to the highest frequencies which have been readily available. Fever machines (for induction therapy), an injection molding machine in which the plastic charge is rapidly and thoroughly heated by induction, and a proposal for the rapid and distortionless drying of green ceramic bodies are examples. Such heating is expensive, but in special cases the cost is reasonably low for the advantages obtained. Active interest in this field suggests much to be done with electrons dancing to the beat of the rhumbatron.

WEIGHT OF STORED ENERGY.—An abstract appearing in Mechanical Engineering, November, 1939, published by the American Society of Mechanical Engineers, 29 West 39th Street, New York, N. Y.

An article by Vernon H. Grant, Bureau of Aeronautics, Navy Department, appearing in Product Engineering for September, 1939, attempts to evaluate the various forms of stored energy in terms of weight in connection with

WEIGHT-ENERGY CONVERSION FACTORS

Type	Weight, lb per kwhr	Container weight, lb per kwhr	Total weight, lb per kwhr
Gasoline	0.18	0.02	0.20
H_2 gas sphere	0.06	1.92	1.98
H_2 gas cylinder	0.06	2.97	3.03
H_2 liquid	0.06	0.12	0.18
Cartridge	2.78	2.78	5.56
Dry battery	45–65	...	45–65
Storage battery (10 hr)	108	...	108
Storage battery (3 min)	480	...	480
Compressed air (100 psi)	281	600[1]	881
Compressed air (500 psi)	204	420[1]	624
Compressed air (1000 psi)	128	273[1]	401
Compressed air (5000 psi)	93	200[1]	293
Flywheel (not including hub and web)	302	...	302
Rubber	1290	...	1290
Steel, tension or compression	147500	...	147500
Steel, spring	443000	...	443000

[1] Container weights based upon spheres, cylinders are 1 1/3 to 2 times as heavy.

the study of auxiliary power in airplanes. The author says that suggestions often advanced involving energy storage, while possibly not completely fantastic, are still so far from the optimum designs obtainable that a fresh start, based upon more sound concepts of relative weights, is usually necessary. It is with the latter object in mind that he has studied the different sources of power and compiled the following table in terms of the electrical unit pounds per kilowatt hour, since electricity is usually considered as the primary means for distributing auxiliary power.

STATIC ELECTRIC HAZARDS ON BOARD OIL TANKERS.—An article by George F. Prussing, Safety Engineer, Union Oil Company of California. Reprinted in the July, 1939, issue of the Pacific Marine Review, San Francisco, Calif. Originally published in the May, 1939, issue of Marine Safety.

When any liquid flows through pipes or hose or through the air from one vessel to another, static electricity is generated. This in itself is not dangerous unless it can accumulate on some object that is thoroughly insulated from the ground or from the vessel from which the liquid is being discharged. If the liquid is a conductor of electricity, such as water, there is little chance of accumulation. Petroleum being a non-conducting liquid does not permit the charges to escape through it. If accumulation takes place on an insulated object, such as a bucket or tank being filled, or even a bucket floating on the surface of a tank, this accumulation becomes hazardous in the presence of an explosive mixture when the accumulated static charge is allowed to jump as a spark to some other object.

On board ship, tanks are usually filled by means of pipes which enter the ship's tanks at or close to the bottom. Overall loading of materials which give off flammable vapors, that is, those which have a flashpoint below 100 degrees, is forbidden. There is no possibility of accumulating a static charge as the result of loading a tankship when our rules are followed, since the ship's piping, the tank walls and the hull of the ship are metallically one, and the ship is resting in water, which perfectly grounds it.

In the case of the pipe on the wharf the static generated is not hazardous because the pipe is grounded either by resting on and through the earth or by means of attached bonding wires which are carried to salt water under the wharf. Hence the charges do not accumulate. In the case of the hose, accumulation of static can occur on hose couplings if they are not connected to the metallic reinforcing of the hose and thus bonded either to the ship or to the pipeline. Some use is made in the oil business of hose which carries no metal reinforcing and no bond wires and in cases of this sort a coupling supported between two lengths of hose is completely insulated and may accumulate a considerable charge of static electricity. This charge has been felt by men who have come in contact with couplings at a time that a spark is drawn from them, a fire may result. This type of hose should never be used at any time during the loading or discharging of Class A oils. The hazard may be eliminated by the use of bond wires in hose if the wire is properly connected to the nipple at each end.

No description of the static hazard is complete without pointing out the difference between it and the stray currents arising from power circuits on land or on the wharf which sometimes leak onto wharf piping and consequently are transmitted to the hose couplings through spiral lining or bond wire. The two following examples illustrate how this may occur:

1. When one of our ships was connected to the pipe stem on the Tacoma smelter wharf several years ago, a spark was seen to jump from the pipe flange to the hose flange. The hose was already connected to the ship's piping. Investigation revealed that the little electric railway used by the smelter was not properly protected against leakage of the return current through the rails and this current got into the piping on the wharf. This piping was not properly grounded. When the ship's hose was first made fast to the ship's piping, and then connected to the wharf piping, some of this current sparked across the two flanges seeking the ground through the ship. No fire resulted because the hose contained no flammable vapors. The cure for the situation was found by inserting an insulating flange in the pipeline near the hose connection. The remainder of the pipeline was then permanently grounded to salt water.

2. At one of our marine plants a power circuit running in conduit under ground failed and the conduit itself became charged with the current. The conduit was grounded by connection to wharf piping, consequently the wharf piping also became alive. Because the ground was dry sand, not enough current leaked off to cause the heavy fuses in the power circuit to blow out. When the tanker bond wire was next used, a flash occurred when the wire was thrown to the deck of the tanker. The wire had already been connected to the outlet on the wharf, which in turn was connected to the pipeline. This method of connecting the bond wire was in direct violation of company orders, which state that the connection shall first be made to the ship and then to the wharf receptacle, which is arc tight and consequently safe from sparks. In this case, however, sparks jumped from the end of the wire when it was thrown on the deck of the tanker, and had explosive vapors been present a fire would have resulted. Proper grounding of the wharf piping would have revealed the defect in the power circuit by blowing the fuses.

The use of a bonding cable between ship and shore is solely for the purpose of eliminating fires which may result from incidents such as those mentioned above. It has no function as a preventive of static, since both ship and piping are assumed to be properly grounded. There is no question but what the ships are always properly grounded, but since the short piping sometimes gets a charge of shore current the bond wire is used in order to assure that the piping is grounded through the ship before hoses are connected. It is obvious that the connection must first be made to the ship so that if a spark occurs it will be in the arc-tight receptacle on the wharf, where it can do no harm.

Shore accidents resulting from static are very few. At sea and on the waterfront they are almost unknown. The principal trouble resulting from this phenomenon is in refineries under conditions that do not prevail anywhere else in the oil business.

REVIEW.

THE POSSIBILITIES OF THE CONSTANT PRESSURE GAS TURBINE (French), Bulletin de l'Association Technique Maritime et Aéronautique, Paris, 1939, pp. 205-230.

The paper under the above title was presented by M. E. Gautier, Captain of the French Engineer Corps, now in the Ministry of Marine. It is a very well written analysis of the possibilities of gas turbines for marine propulsion. The author analyzes the merits of the constant volume cycle and the constant pressure cycle and indicates that the latter has the greater advantages for marine use. He indicates that blading can now be used at an average temperature of 1112 degrees F. and advocates a pressure of 128 pounds per square inch and a peripheral velocity of not over 1082 feet per second. A particularly interesting contribution is the author's recommendation that the gas turbine be built with a single disc somewhat after the manner of the Terry steam turbines or the Westinghouse return turbines wherein the steam made several expansions through the same disc by means of return chambers. The primary purpose of this recommendation for gas turbines is that a very high initial temperature could be used since the blading on the disc would assume a temperature intermediate between the initial temperature and the final discharge temperature.

Detailed formulas are given for the computations necessary in connection with a gas turbine and a number of results are presented. Basic data is presented for a 40,000 HP. installation for which fuel consumption is predicted to be 0.46 pound of oil per horsepower hour (with oil at 18,500 Btu. per pound).

BOOK REVIEW.

THE JUDGING OF THE STABILITY OF SHIPS AND THE DETERMINATION OF THE MINIMUM AMOUNT OF STABILITY. By Jaakkd Rohola, Published in Helsinki, Finland, Price, 100 Finnmarks. Reviewed by J. C. Niedermair, Principal Engineer, Bureau of Construction and Repair, Navy Department, Washington, D. C.

The data collected in the book and the technical conclusions are intended for use in connection with ships operating in Finnish waters. The subject is treated in an exhaustive manner. The historical data assembled by the author includes valuable information concerning many vessels which capsized or were lost at sea. Unfortunately the text is a rather difficult one to understand in spots. This is owing to the translation from original Finnish to English text. Nevertheless, it is an excellent and valuable study of a difficult and controversial subject. The scope of the subject matter covered in the text is indicated by the several chapter headings.

Chapter I—The Phases in the Development of the Problem Concerning the Minimum Amount of Stability.

Chapter II—Foundations for Judging the Minimum Stability.

Chapter III—Main Principles of the Methods for Judging the Stability of Our Seagoing Vessels and Those on Our Waters.

Chapter IV—The Judging Method for the Stability of Our Seagoing Vessels.

Chapter V—The Judging Method for the Stability of Our Vessels on the Inner Waters.

Chapter VI—Numerical Examples of the Application of the Judging Methods Explained.

ERRATA.

An article, *Calorific Value-Gravity Relationship of Fuel Oils,* by Cyrus Beck, appeared in the November, 1939, issue of the JOURNAL. A check of the computations and derivation of the formula for the subject relation has disclosed several arithmetical errors.

Table 1: Column 2, line 5, should read S-1533 instead of S-1535. Heading of column 8 should read N.B.L. No. instead of B.T.U. per lb.

B.T.U. per lb. of fuel oil, reference number 195, should be 18309.

Recalculation shows that the following values are correct for the symbols shown at the bottom of page 566:

N	= 266	$(X')^2$	= 3921
(X′)	= —725	$(Y')^2$	= 3552
(Y′)	= —512	(X′Y′)	= 3368

Substitution of these values in the equations listed gives the following results:

(Final Equation)	$Y = 17770 + 50.7X$	(Equation 7)
(Standard Error of Estimate)	$S = 146$ B.T.U.	(Equation 10)
(Coefficient of Correlation)	$r = 0.883$	(Equation 8)
(Probable Error of Coefficient of Correlation)	P.E. $= \pm .009$	(Equation 9)

Correction of the data given in Tables 3 and 4 and replotting of the curves of Figures 1, 2, 3 to conform with the above equation requires amendments to the following paragraphs or subparagraphs:

Page 569, last paragraph—In the first and second lines, change " smaller than " to " between." Delete second sentence.

Page 570, Subparagraph (c)—Change " increase " to " decrease." Subparagraph (e)—Insert the word " Most " before " Bunker ' B ' oils, etc."

The correct Tables 3 and 4, and the correct curves of Figures 1, 2, 3 follow:

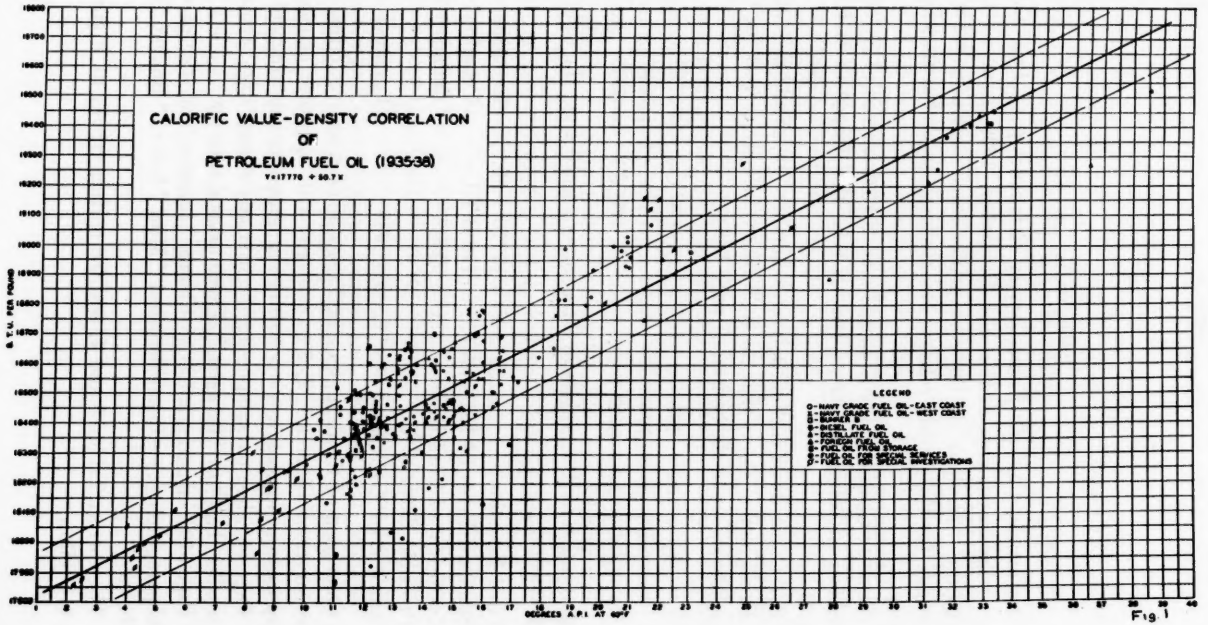

CALORIFIC VALUE-DENSITY CORRELATION
OF
PETROLEUM FUEL OIL (193538)
V=17770 + 50.7 x
B.T.U. PER POUND
DEGREES A.P.I. AT 60°F
LEGEND
D-NAVY GRADE FUEL OIL-EAST COAST
E-NAVY GRADE FUEL OIL-WEST COAST
B-BUNKER B
G-DIESEL FUEL OIL
A-DISTILLATE FUEL OIL
C-MOTOR FUEL OIL
F-FUEL OIL FOR STORAGE
H-FUEL OIL FOR SPECIAL SERVICES
I-FUEL OIL FOR SPECIAL INVESTIGATIONS

Fig 1

TABLE 3.

Year	*Formula*	*10° API*	*20° API*	*30° API*	*Coefficient of Correlation*	*Standard Error*
1933	$Y = 17687 + 57.9\ X$	18,266	18,845	19,424	$.912 \pm .016$	128 BTU
1935	$Y = 17751 + 53.4\ X$	18,285	18,819	19,353	$.842 \pm .028$	114 BTU
1938	$Y = 17770 + 50.7\ X$	18,277	18,784	19,291	$.883 \pm .009$	146 BTU

TABLE 4.

Formula	*Fuel Oils*	*10° API*	*20° API*	*30° API*
$Y = 17010 + 90\ X$	Straight run	17,910	18,810	19,710
$Y = 17645 + 54\ X$	Cracked	18,185	18,725	19,265
$Y = 17770 + 50.7\ X$	N.B.L. (1938)	18,277	18,784	19,291

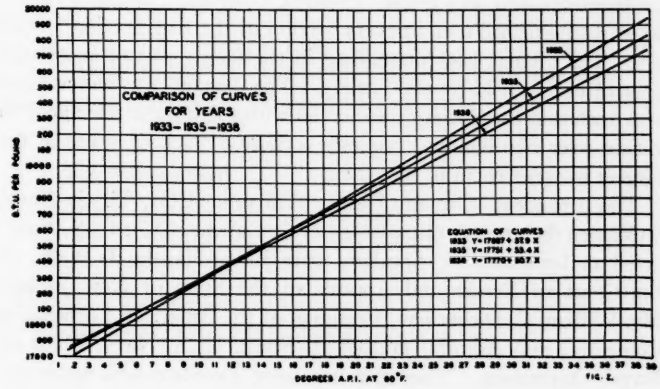


FIG. 2.

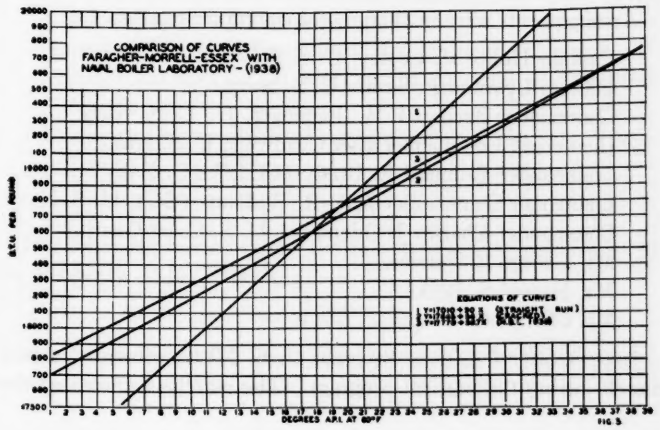


FIG. 3

ASSOCIATION NOTES.

ANNUAL BANQUET

The Annual Banquet of the Society was held at the Willard Hotel at Washington, D. C., on Friday, April 5, 1940.

Approximately 975 guests attended, comprising a group of the most distinguished men in every section of the Engineering Industry. The dinner was outstanding in the long series of most successful events held by the Society. The demand for reservations was such that the Society was forced with the utmost reluctance to return over two hundred requests because of lack of space in the largest banquet room in the city.

The President, Captain Henry Williams (CC), U. S. Navy, welcomed the guests. Captain James O. Gawne (CC), U. S. Navy, acted as Toastmaster. Speakers were Judge Harold B. Wells, of the Court of Errors and Appeals of the State of New Jersey; Mr. William C. Dickerman, President of the American Locomotive Company; and Admiral Harold R. Stark, U. S. Navy, Chief of Naval Operations. The speeches were excellent and were received with much enthusiasm.

Arrangements were in charge of a committee composed of:

Captain Hollis M. Cooley, U. S. Navy.
Captain Lybrand P. Smith, U. S. Navy.
Commander Frederick W. Pennoyer, Jr., U. S. Navy.
Lieutenant Commander Guy Chadwick, U. S. Navy.
Commander Edmund E. Brady (CC), U. S. Navy.
Lieutenant Commander Milo R. Williams (CC), U. S. Navy.
Commander Charles J. Odend'hal, U. S. Coast Guard.
Mr. George E. Ferrier.
Mr. Harold S. Beck.
Mr. Charles G. Cooper.

Lieutenant Charles Benter, U. S. Navy, presented a new march during the musical program. The march is known as the

American Society of Naval Engineers March and is dedicated to the Society. It was played for the first time at the banquet by the Navy Band.

The Society desires to express its appreciation to Lieutenant Benter for his long and successful service in providing musical entertainment at successive annual banquets since the year 1919.

Charles Benter was born April 29, 1887, in New York City. He enlisted in the Navy March 20, 1905. He was designated as Leader of the U. S. Navy Band March 10, 1925, with the pay and allowance of a Lieutenant, U. S. Navy, and authorized to wear the uniform of a Lieutenant, with a musician's lyre as a Corps device. He was commissioned as a Lieutenant June 7, 1935, by Act of June 7, 1935, Public No. 113—74th Congress. He was the first Bandmaster in the U. S. Navy to be so honored.

Medals:

Cuban Pacification.
Mexico.
World War.
Good Conduct.

Foreign Decorations:

Order of St. Sava (II), Jugoslavia.

Memberships:

Army and Navy Club, Washington, D. C.
National Press Club, Washington, D. C.
American Society of Composers, Authors and Publishers.
Member and ex-Vice President of the American Bandmasters Association.
Honorary Degree of Mus.D. by Columbus University, Washington, D. C., 1929.

MEMBERSHIP.

The following having joined the Society since the publication of the February, 1940, JOURNAL:

NAVAL.

Alexander, Robert T., Lieutenant, U. S. C. G.
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Holden, William B., Lieutenant Commander, U. S. N.

Klakring, Thomas B., Lieutenant, U. S. N.

Lee, Carl F., Machinist, U. S. N.

McIntosh, James, Ensign, U. S. C. G.

McQuiston, Irving M., Lieutenant Commander, U. S. N. R., Bureau of Aeronautics, Navy Dept., Washington, D. C.

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Wagline, J. H., Lieutenant, U. S. C. G.

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Webster, Hugh P., Lieutenant, U. S. N.

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ERRATA.

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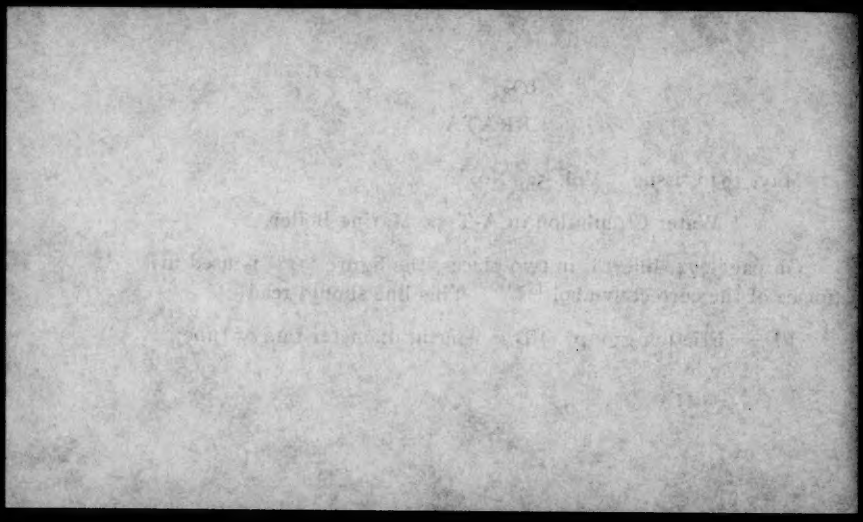
"Water Circulation in A-Type Marine Boiler."

On page 274, line 18, in two places, the figure "1" is used in place of the correct symbol "l". This line should read

"fl/D = Friction groups (l/D = length, diameter rate of tube;"